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SPIRALFLOW 2: Development and demonstration of a prototype for decentralised ventilation with spiral heat exchanger

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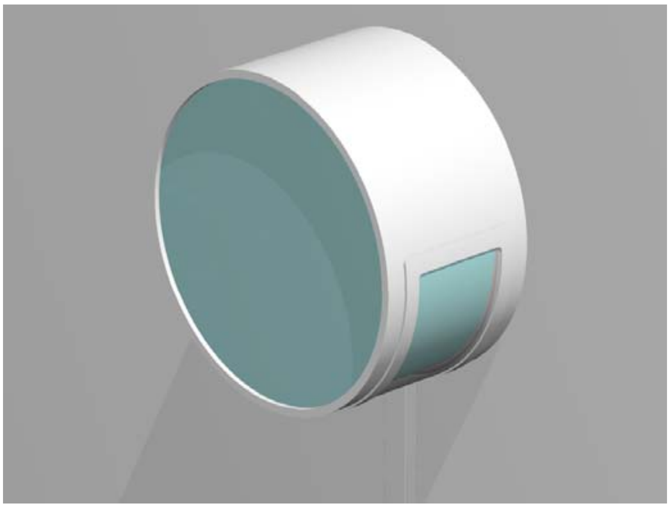


SPIRALFLOW 2

Projekt nummer 348-046

Udvikling og demonstration af en samlet prototype til decentral ventilation med spiralformet varmeveksler

ELFORSK

PLH arkitekter

BRØNNUM PLAST

DTU Civil Engineering
Department of Civil Engineering

Smith
Innovating construction

ebmpapst

SUSTAIN SOLUTIONS

DTU Elektro
Institut for Elektroteknologi

Table of content

TABLE OF CONTENT 1
FOREWORD 3
RESUME (DANISH) 4
Formål 4
Resultater 4
RESUME (ENGLISH) 6
Aim of the project 6
Results 6
BASIS FOR THE PROJECT 8
INTERACTION WITH OTHER FUNDED PROJECTS 9
THE AIM OF THE PROJECT 10
Vision 10
The challenge 10
The means to get there 10
RESULTS 11
Prototype 11
Design 11
Production 19
Predicted performance 24
Filters 24
Heat exchanger 24
Fans 26
Bypass 26
Experimental setups and measurements 27
Heat recovery, air flow and power consumption 27
Air flow and Power consumption 30
Leakage 32
Acoustics 33
Market analysis 37
Data og sources 37
Results 37
Estimated sales 38
General assumptions: 38
Scenario 1 and 2 (Small scale) 38
Scenario 3 and 4 (Large scale) 38
Results 39
DISSEMINATION 40
DISCUSSION 41
Test results 41
Future market 41
CONCLUSION 43
PERSPECTIVE 44
Future development and production of Spiralflow 44

APPENDIX A – TRANSMISSION NOISE OF BREATHE55 45

APPENDIX B – TRANSMISSION NOISE OF SPIRALFLOW 46

APPENDIX C – SOUND POWER OF SPIRALFLOW 47

APPENDIX D – OVERVIEW OF THE MARKET ESTIMATION TOOL 49

Market example 49
Input of variables 49
Market numbers 50

Foreword

Project '348-046 Udvikling og demonstration af en samlet prototype til decentral ventilation med spiralformet varmeveksler' generally known as *Spiralflow 2*, started in July 2016 and ended in May 2018. ELFORSK has funded the project with 1.114.661 DKK while the partners have contributed with equity contribution of 1.125.032 DKK.

The project team is grateful for the financial, commercial and personal support from Elforsk. A special thanks to Ditte, Jørn, Dorthe and Richard.

During the project, several people have contributed to the development. A big acknowledgement to project participants: Henning Solfeldt (PLH architects), Hasse Brønnum (Brønnum Plast), Torben Lintrup Kirkholt (Ebmpapst), Svend Svendsen (DTU Civil engineering), Kevin Michael Smith, (DTU Civil engineering), Finn T. Agerkvist (DTU Electro), Cheol-ho Jeong (DTU Electro), Mikkel A. Thomassen (Smith Innovation) Charlotte K. Larsen (Sustain Solutions) and Anders L. Jansen (Sustain Solutions).

A special thanks to Jakob Thomsen and Carina Lindahl who based their master thesis in Design and Innovation (DTU) on Spiralflow. With Kevin Michael Smith as a co-suporvisor and Tim McAloone as the main supervisor. Congratulation on the graduation and the grade 12.

This end report is in English. A short resume is provided in Danish with the aim and results of the project. The Danish resume is in the beginning of the report.

Resume (Danish)

Formål

Ved nybyggeri og renovering tætnes boliger i en sådan grad at mekanisk ventilation bliver kritisk for at sikre et sundt indeklima og en sund bygningsmasse. Projektets formål er at udvikle og teste en samlet decentral ventilationsenhed med en nyudviklet effektiv modstrømsveksler som kerneteknologi. Modstrømsveksleren er blevet udviklet i projektet '346-036 Novel spiral shaped counter flow heat exchanger for decentral ventilation' også kendt som Spiralflow 1, som er forgængeren til dette projekt.

Det er valgt at udvikle en enhed med en modstrømsveksler, for at kunne affugte især vådrum. Denne type enhed er derfor udviklet til køkken og badeværelser, men ikke begrænset til brugen hertil, da den også vil kunne installeres i opholdsrum.

Resultater

Det blev i afslutningen af projektet Spiralflow 1 anbefalet at varmeveksleren skulle videreudvikles, eftersom det meget lave tryktab i varmeveksleren medførte en skæv luftfordeling igennem veksleren. Veksleren er derfor blevet videreudviklet i dette projekt med strategisk placeret modstande, som både sørger for at luftfordelingen forbedres, men også sørger for at afstanden mellem lagene i veksleren holdes præcist. Dette var i Spiralflow 1 en af de største udfordringer.

I Spiralflow 1 blev der produceret vekslerlængder på hhv. 28cm og 34cm. Test heraf viste at, hvis der skal opnås høj nok varmegenvinding, skal veksleren være omkring 34cm. Der er i dette projekt derfor produceret en ny veksler med de indbyggede strategiske modstande i længden 34cm.

Begrænsningen på de 34cm i vekslerlængde gav anlæg til konceptuelle diskussioner om hvordan enheden i sin helhed skulle designes. Eftersom at projektholdet har erfaring fra markedet for decentral ventilation med enheden Breathe55, blev der lagt fokus på at enheden skulle have det mest diskrete udtryk for den udvendige del. Den udvendige del flugter derfor helt med den udvendige del af muren, hvor der kun kan ses en almindelig ventilationsrist. Dette adskiller Spiralflow fra andre enheder i markedet. Det har dog medført at den indvendige del er blevet lidt større ift., hvordan lignende konkurrerende enheder er udformet.

Da enheden baserer sig på en modstrømsveksler, forventes der en del kondensdannelse inde i enheden. Det har fra projektholdets side fra starten af projektet været ambitionen at denne kondens ikke bare skulle have lov til at dryppe til udvendig side, som man kender det fra konkurrerende produkter og f.eks. air condition enheder. Der er derfor udviklet en løsning, hvor kondensvandet opsamles og via en tilkoblet gummislange på enheden føres til det nærmeste afløb. Da hovedfokus for denne type af enhed er badeværelser og køkkener, må det forventes at der vil være afløb placeret tæt på enheden, hvorfor denne løsning er god. Ved placering i opholdsrum vil denne løsning give udfordringer. Derfor er det forberedt i enheden at gummislangen kan føres gennem enheden og til udvendig side, hvor det evt. kondensvand kan dryppe frit. Ved placering i opholdsrum forventes dog væsentligt mindre kondensdannelse, hvorfor dette anses som acceptabelt.

Udover det konceptuelle design er der lavet tekniske design for en række detaljer i enheden, her kan bla. nævnes:

- bypass funktion med tilhørende steppermotor og styring
- kondensdræn
- filterplaceringer, arealer og filterklasser

- hvordan enheden er tænkt serviceret, ved at kunne tage enheden ind og ud af væggen
- Forlængelse af enheden ved variation i vægtykkelsen

Alle disse detaljer har udmøntet sig i en funktionel prototype, som er blevet anvendt for test på DTU.

Sideløbende er der blevet udviklet en styring, som vil kunne kontrollere ventilatorerne og bypass ventilen ud fra input fra temperatur, fugt og CO_2.

Der er produceret én fuld prototype som er blevet testet i laboratorie og et testhus på DTU, samt i et akustiklaboratorie. Et af målene i projektet var at teste enheden i virkelige boliger med beboere der kunne give feedback. Denne del er ikke lykkedes, da der var behov for videreudvikling af varmeveksleren førend at udviklingen af selve prototypen kunne starte. Testhuset på DTU har ladet enheden blive testet i vind og vejr, men ikke med beboere der har kunne give feedback.

Prototypen har fået målt fantastisk ydeevne ift. varmegenvinding og strømforbrug. Begge parametre overholder BR2020 kravene, hvilket er rigtig flot taget i betragtning at dette er en første prototype.

Støjdæmpningen udefra har vist gode resultater set ift. enheder der allerede er på markedet. Spiralflow enheden performede en lille smule dårlige end enhederne på markedet, men prototypen var blot sat i et hul i en væg i akustikkammeret og ikke installeret med fugemasse og lignende, som færdigproducerede enheder gør. Det forventes derved at støjdæmpningen udefra vil være helt på linje med allerede eksisterende enheder.

Resultaterne for støjen som selve enheden generer, var ikke så positive som håbet. Årsagen skyldes ventilatorerne. I den nuværende prototype er der valgt to forskellige ventilatorer. Den udvendige ventilator er ikke opdateret ift. en støjreducerende software, da producenten ikke har denne klar endnu. Dette forventes at forbedre støjresultaterne. Efter støjmålingerne blev ventilatorplaceringen undersøgt. Det afslørede at ventilatorerne ikke var placeret optimalt ift. ydeevne og derved støj. En optimal placering forventes at forbedre støjresultaterne.

Kortslutningen af luft på både ude og inde er for stor. Dette skal forbedres i fremtiden. På den udvendige side skal en bedre adskillelse mellem de to luftstrømme sikre at mindre luft kortslutter. På den indvendige side kan et mindre filterareal øge hastigheden og derved sikre en bedre adskillelse af de to luftstrømme. Dette vil dog medføre et øget tryktab og derved strømforbruget.

Den tekniske performance har vist sig god for den første prototype, og med allerede kendte forbedringsmuligheder er potentialet stort for en sådan type enhed. Der er dog stadig en række ubekendte faktorer især omkring produktionen af varmeveksleren, som skal afsøges nærmere inden det kan besluttes, hvad den videre færd for produktet vil være. Ligesom en brugerundersøgelse skal laves ift. om den større indvendige del kan accepteres.

Markedsanalysen har vist at den nuværende længde af enheden er et problem ift. det potentielle marked. Hvis enheden kunne laves kortere så den kunne installeres i tyndere vægge vil det potentielle marked i Danmark stige.

Resume (English)

Aim of the project

Mechanical ventilation is becoming more and more a necessity in residential homes in order to secure a healthy indoor environment. When facades are air tightened in order to save energy for heating a need for increased ventilation arisen, in order to secure a good air change which ensures a healthy environment. Residents are commonly known for not being good enough to open the windows. Humidity and particles are then accumulating in the indoor air. This provides humid living spaces and can lead to poor health conditions such as asthma, due to the growth of fungus.

Many residential homes do not have mechanical ventilation with heat recovery. Moreover, it is often expensive to install in existing buildings. There is therefore a market for room-based decentral ventilation units with easy installation. These units can ensure demand controlled mechanical ventilation with a high rate of heat recovery and a low power consumption. At the same time, they allow the possibility of venting when overheating occur.

The *Spiralflow* 2 project is about creating a new decentral ventilation unit with a new type of counterflow heat exchanger, which was developed in *Spiralflow 1*. The use of a counterflow heat exchanger ensures a high moisture removal potential which makes it possible to install the unit in bathrooms and kitchens. However, it is not limited to kitchen and bathrooms.

Results

At the end of the project *Spiralflow 1* it was recommend that the heat exchanger should be further developed. As the air distribution through the heat exchanger was uneven due to the very low pressure drop. In *Spiralflow 2* the heat exchanger was optimized with strategic placed air resistances which improved the air distribution. As a second benefit the resistances also acts as a spacer material which keeps the distance between the two plastic sheets in the heat exchanger. This was one of the main obstacles in *Spiralflow 1*.

Tests in *Spiralflow 1* had revealed that the optimal heat exchanger length was around 34cm. In *Spiralflow 2* a 34 cm long heat exchanger with the resistances has been constructed.

From the limitation of the length of the heat exchanger arose a conceptual discussion of how a complete unit should be designed. By market experience from decentral ventilation units from the Breathe55 unit, it is known that especially architects are more concerned by the exterior side than the interior. It was therefore decided to make a unit with an exterior lid that is completely flush with the façade. This separates Spiralflow from other units in the market. The cost of this is that the interior part then is larger than competitors.

Condensation is expected in the unit as it has a counterflow heat exchanger. The project team has from the beginning had the ambition that condensation shouldn't be allowed to drip freely to the outside. This is known from competitor products and e.g. air conditioners. A solution where the condensation is then drained by a rubber hose to the nearest drainage was therefore developed. The main focus of the Spiralflow unit is bathrooms and kitchens and a drainage near by the unit is therefore expected. Installation in dry rooms would be challenging for this kind of solution. The unit is therefore prepared to allow the rubber hose to the exterior side and let the condensation drip freely. In dry rooms a lot less condensation is expected and the level of dripping condensation water is therefore believed acceptable.

Besides the overall design of the unit has there have been designed technical solutions for e.g.:

- bypass function with stepper motor
- Condensation drainage
- Filter placement, areas and filter classifications
- How the unit is thought serviced from the inside by taking the unit in and out of the wall
- Extension of the unit for variation in wall thicknesses.

All these details have resulted in a functional prototype used for tests at DTU.

The development of the controls was made in parallel. The controls can control the fans and the bypass valve based on inputs from temperature, humidity and CO_2 sensors.

There has been produced one full prototype used for tests in a laboratory and a test house and an acoustic chamber at DTU. An objective in the project was to test the unit in actual buildings including feedback from the occupants. This part has not been successful. The test house at DTU has allowed the unit to be tested in actual weather conditions, but without user feedback.

The prototype has shown great performance on heat recovery and power consumption, with results greater than the requirements for the 2020 Danish building regulations. This is impressive for a first prototype.

The transmission noise dampening results show good performance compared to other market available solutions. The results are currently a bit weaker than the market, but this could be made up by a complete installation with sealant. The prototype was just installed in a hole in the acoustic chamber without sealants.

Noise generation was not as positive as the unit currently is a bit too noisy. This is due to the fans. In the current prototype the exterior fan is not yet updated with a software which should increase its noise performance. After the measurements and investigation of the fans revealed that their placement wasn't optimal in regard to performance and thereby noise. An adjustment of this is expected to improve the acoustic results.

On both the interior and exterior lids of the unit was there measured too high short circuiting of air. This is a part which needs to be improved in the future. On the exterior side a greater separation of the two air streams is needed. And on the interior side a downsizing in filter area could increase velocities and thereby ensure separation between the two air streams. This will have a negative effect on the pressure drop and thereby the power consumption.

The technical performance from the first prototype has been good. And with already known improvement possibilities is the potential for the Spiralflow unit great. There are still unknowns regarding production of the heat exchanger which should be investigated further. It will also be a good idea to perform a user survey in order to reveal if the large interior lid can be accepted.

A market analysis has shown that the current length of the unit is problem for potential sales. If the unit could be made shorter it could adapt to slimmer walls than 390mm which it currently can adapt to. If so, the Danish market would increase.

Basis for the project

To meet the future requirements regarding energy consumption and indoor climate in both existing and new residences, there is a need for mechanical ventilation with a high rate of heat recovery and a low electricity consumption and with the possibility of venting for too high indoor temperatures. There is a very large, and rapidly growing, need for performing energy renovations of existing residential buildings in order to reach the goal, that all buildings in the year 2035 should be free of fossil fuels.

In older multi-story buildings, it will be ideal to replace the old and poorly insulating and leaky windows with new highly insulated and airtight windows. At the same time there will be a need for installing mechanical ventilation with heat recovery to ensure a good air quality without draught and with a large energy saving. This mechanical ventilation can be a central ventilation system. However, it is often difficult and expensive to install in existing multi-storey residential buildings. Therefore, is there a need for decentral ventilation units, which can be mounted in the exterior wall in each room and deliver balanced and demand controlled mechanical ventilation with high heat recovery and low power consumption.

The decentral ventilation units need to have great acoustic performance. Both regarding noise generated by the fans, but also from noise generated outside the building, e.g. traffic noise. In practice the occupant should not be able to hear that a decentral ventilation unit has been installed.

The airflows in the room-based decentral ventilation units can be regulated by changing the speed of the fans and thereby making it easy and inexpensive to exploit a demand-controlled ventilation strategy, where flow is adapted to the activity level in each room. Research has unveiled that there is a great need for boosting the ventilation level in bedrooms. At the same time, many Danish people prefer a cooler temperature in the bedrooms than the usual 20-22°C in the rest of the dwelling. These needs can be fulfilled by the use of a room-based decentral unit which ventilates the bedroom independent from the other rooms.

Interaction with other funded projects

The project team has in the former project “Decentral Komfort”, funded by Fornyelsesfonden and later Markedsmodningsfonden, worked with developing and maturing a whole decentral ventilation unit. This has resulted in the fully developed and market available unit Breathe55. Breathe55 is based on a rotary heat exchanger, which gives great heat recovery and control options and allows the unit to be installed in slim walls down to 250mm in thickness. Which is market leading. The rotary heat exchanger however has a limit for the drying potential and is therefore intended for dry rooms.

The predecessor project for this project has been funded by Elforsk ‘346-036 Novel spiral shaped counter flow heat exchanger for decentral ventilation’, generally known as Spiralflow 1. In Spiralflow 1 the new counter flow heat exchanger was developed. Spiralflow 1 is therefore the basis for Spiralflow 2, which uses the newly developed heat exchanger as the core technology in a complete decentral ventilation unit.

The aim of the project

Vision

The overall vision is to realize energy savings and improve the indoor climate by creating an installation friendly industrial product which sets market standards in relation to indoor air quality, thermal comfort, low energy consumption, efficiency and low transmission noise.

The vision is to make the product as widely applicable as possible in relation to building typology and installation process, so the international market potential can be exploited.

The challenge

The coming legislations regarding energy- and indoor climate for new housing combined with an urgent need to conduct energy-refurbishments in existing residential buildings creates big challenges in relation to securing a good indoor climate, having low total costs and a low energy consumption.

In this project the newly developed counter flow heat exchanger from the previous project has been built into a complete decentral ventilation unit. The prototype has shown that it is possible to overcome these new legislations regarding high heat recovery and low power consumption.

The means to get there

DTU Civil, DTU Electro, EBMPapst, Brønnum Plast, PLH Arkitekter and Sustain Solutions has optimized the spiral shaped heat exchanger developed in the previous project. And built the new heat exchanger into a functioning prototype with fans, lids, condensation drainage, bypass valve and filters.

The prototype has shown promising results in regard of thermal heat exchanging efficiency, power consumption and transmission noise.

The material costs for the heat exchanger are low, but the production cost has not yet been determined. As this is a new type of heat exchanger the production of it needs to be investigated further. This will have a great impact on the final production cost.

Results

Prototype

Design

Prerequisites and process

Size matters! Almost all functional parameters in the Spiralflow 2 ventilation unit will improve by increasing the unit's diameter and length, while the visual and physical impact on the building and its rooms will suffer from it. That is why the design for the prototype shall balance between functionality and size.

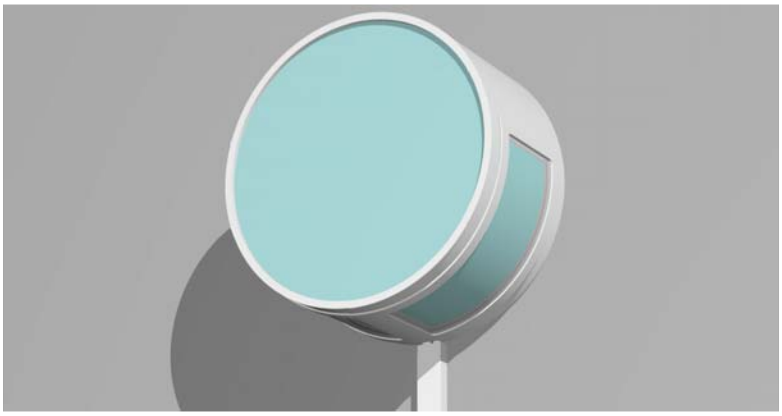


Figure 1 - Internal lid of the Spiralflow unit.

On an early stage, we decided that the Spiralflow unit shall fit into Ø255 mm wall holes as the Breathe 55 unit does. The model of the heat exchanger in Spiralflow 1 is Ø250 x340 mm and a small adjustment of the diameter to Ø240 mm makes it fit within the Ø255 mm hole in the wall without a critical reduction of its energy efficiency.

The rotating heat exchanger in the Breathe 55 unit is only 150 mm long, which allows it to fit in a 250 mm thick wall when its fans are located inside the lids on both sides of the wall. Our Spiralflow heat exchanger is 340 mm, which means deeper lids (more protrusion from the wall surfaces) and/or limitation to use in thicker walls.

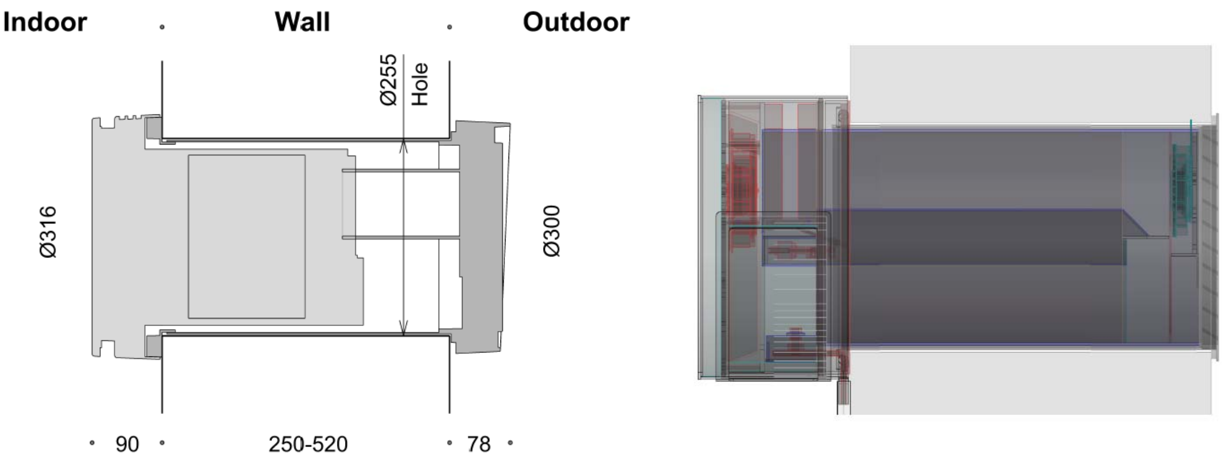


Figure 2 - Cross section of the two decentral ventilation units, Breathe55 on the left and Spiralflow on the right.

From the Breathe 55 unit we have experienced that architects care more about minimizing the visual impact of the unit on the façade than on the rooms. Therefore, we decide to keep the Spiralflow 2 prototype almost flush with the facade. We also choose the minimum wall thickness to be 390 mm and we accept a 180 mm deep internal lid to adapt the total minimum length of the prototype. We also design a concept for extension the unit to fit in with lager wall thicknesses than 390 mm.

To reduce noise from fans and airflows we decide to use rather large filter areas, which leads to smaller pressure losses, lower fan speeds and less power consumption. The price for these functional benefits is to accept a rather large lid (approx. Ø317x180 mm) on the inside of the wall.

Spiralflow is a counterflow heat exchanger and therefore condense of water will occur. Most competitive products drip the water to the outside, but we decide to drain to the inside. Our focus is to use Spiralflow in bathrooms and kitchens were drainage installations are available. Spiralflow units installed in living rooms and bedrooms will need installation of a small pipe to drain the condense water or alternatively a pipe in the bottom of the unit to drain to the outside.

A bypass function shall prevent freezing temperatures in the heat exchanger during cold winter and allow inlet air to pass by the heat exchanger during warm summer. The center tube of the heat exchanger naturally gives the bypass duct.

On the outside, a thin F2 filter protects the heat exchanger from dust in the inlet air. The F2 filter needs to be replaced twice a year, and to get fast access to the heat exchangers external side, the unit shall easily be drawn into the room and after filter change pushed back again to its position in the wall.

Design of prototype

We design the prototype based on the prerequisites mentioned above. Furthermore, we try to add some practical functionalities and esthetic qualities to the visual parts. Never the less, it is a functional prototype and not a fully developed and final product.

Figure 3 - Image of the interior side of the prototype being inserted into a wall.

Mounting the unit

1. Wall tightening. We fix a standard PVC tube - Ø250 mm – to a standard external ventilation grill. The tube is cut in length to fit the actual wall thickness and positioned from outside into the Ø255 mm hole in the wall. A tightening profile between the grill and the wall prevents air and water to penetrate the hole in the wall from the outside. From the inside, we fixate a ring to the wall by silicone and we screw the tube to the wall ring from inside in radial directions.
2. We push the unit into the wall tube until it hits the inside of the grill. A kind of spring might keep the unit in place with some friction. (This detail is not developed.)
3. The drainage pipe fixed to the bottom of the wall ring penetrates a small water basin trough a rubber bushing in the unit when we push the unit in place.
4. The power cable enters the wall ring from above or underneath. We plug in an electrical connector between the wall ring and the unit. Finally, we push a cylindrical cap in place from the front (fixed by friction).
5. When changing filters, we pull off the cap to get access to the M5 and G4 filters on the inside, and we unplug the power cable and draw out the unit to access the external F2 filter.

Heat exchanger

We have two major challenges when designing and producing the heat exchanger. The first is how to line up and secure approx. 2 mm distance between the two foils, and the second is how to seal easily every second layer at each end.

It is possible to imprint the foil with small bumps to secure the distance to the next foil layer, and we hope that these bumps can be imprinted mass production wise. This seems to be a great idea, but if requires further development and some investments to be realized.

For the prototype, we add self-adhesive bumps in a certain pattern to the two foils. We design the pattern to prevent bumps in the one layer to step on bumps in the second layer. The pattern is the same on both foils, but we turn the one foil and use its end as start, before vining them up.

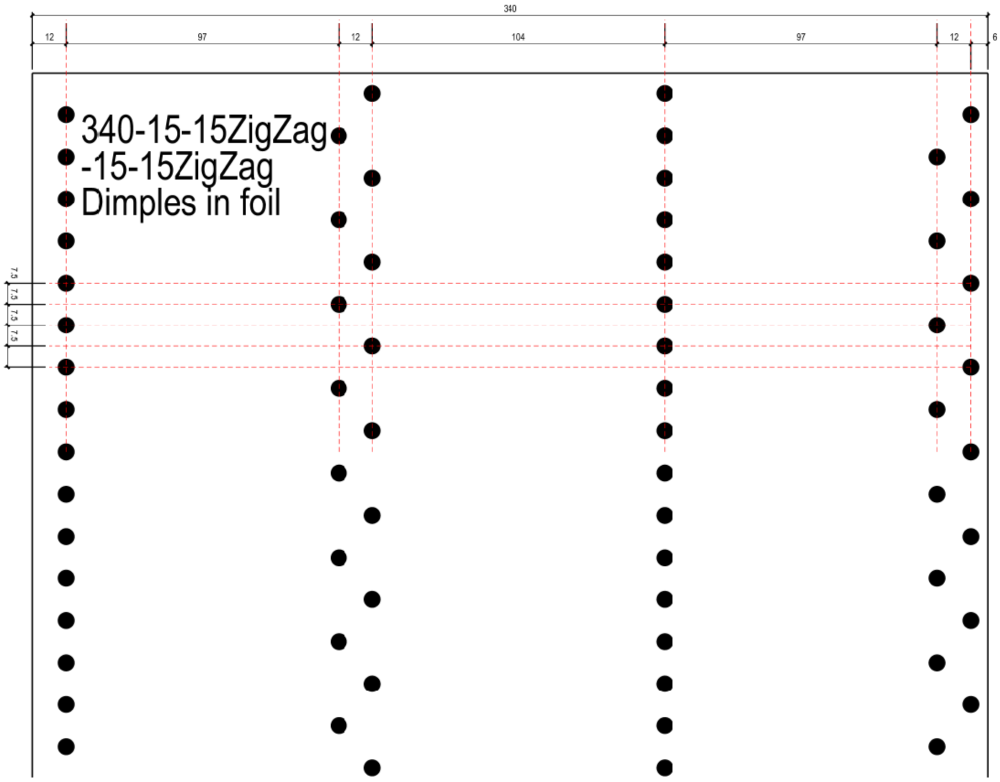


Figure 4 - Pattern used for the placement of bump-ons ensuring an even air distribution through the heat exchanger and a uniform separation between plastic foils.

Figure 5 - Placements of the bump-ons on the plastic foil, before the heat exchanger is rolled up.

The rows of bumps add some pressure loss and make the airflows more evenly distributed over the foil areas. This means that the warm and cold airflows cross each other over larger areas of the foils, and the efficiency of the heat exchanger increases.

In each end of the heat exchanger we seal between every second layer of foil with sealants, but it is too time consuming and manually for mass production. Maybe we can place 2 mm thick self-adhesive tape in right lengths and distances along the side edges of the foils before vining them up. We need to address this issue in the next developing stage.

Airflows and noise

The external fan sucks exhaust air through the rather large G4 filters in the lower half of the internal lid. The exhaust air enters the lower half of the heat exchanger and leaves the upper half on the outside, enters the outside fan and blows out through the top part of the grill in the façade.

The internal fan sucks the inlet air from the lower part of the grill, through the G2 filter into the lower part of the heat exchanger. The inlet air leaves the inside of the heat exchanger from the upper half, enters the fan and blows slowly our through the large M5 filter in front of the lid.

We choose backward curved centrifugal fans from Ebm-Papst on both sides of the heat exchanger. This type of fan push the air out radially into a pressure chamber, followed by rather low noise levels when ideally designed with large chamber volumes. We are able to design a rather large pressure chamber in the inside lid, which is crucial to reduce the noise so close to the users, while the space for the pressure chamber on the outside is very limited.

On both sides of the heat exchanger, we want to use a fan with a 100 mm diameter, which is able to deliver 50-60 m3/h airflow. Unfortunately, the right Ø100 mm fan for the job is still under development at Ebm-Papst, so we have to choose a rather noisy Ø100 mm fan. This fan can be acceptable for the prototype on the outside but not on the inside. Therefore, we choose a larger fan Ø138 mm with a low noise level to the internal pressure chamber although we are have to mount the fan rather close to the upper edge of the chamber. This is the best possible solution for the prototype, but in the next development stage, low-noise Ø100 mm fans will be right to use.

Drainage

Condense water from the exhaust air runs backwards against the flow direction and drips down to the small water basin inside the heat exchanger's indoor lid. From here, it runs through the metal pipe and further into the rubber hose to the drainage installation. A small sensor shall register the water level in the basin, and it sends a signal if the water level rises due to blocking of the rubber hose.

Bypass

The inlet air can bypass through the center tube of the heat exchanger, when the heat exchanger's efficiency shall decrease. The maximal bypass function is relevant during summer, when the cooling effect of the inlet air is appreciated indoor. In wintertime, the temperature of the exhaust air in the heat exchanger can potentially fall to 0° C or lower and the condense water might frieze. If this risk occurs, a smaller part of the inlet air must bypass the heat exchanger to reduce the cooling effect by the inlet air.

We control the bypass function by a damper inside the inner tube and a stepper motor to open the damper. A small spring closes the damper when bypass of inlet air shall stop.

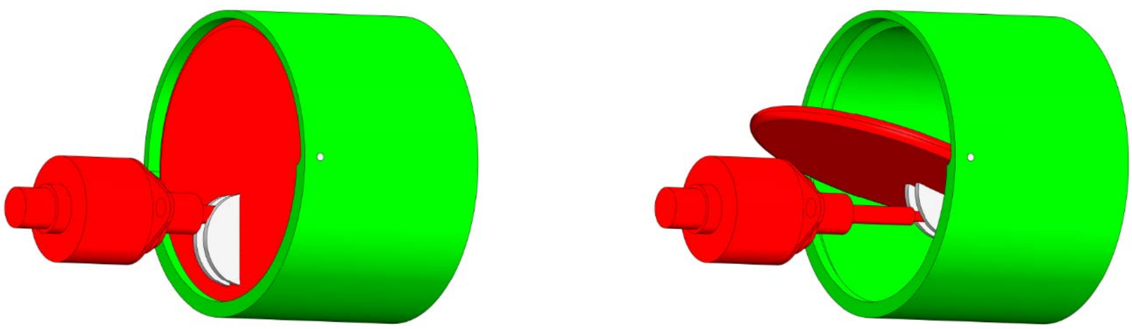


Figure 6 - Drawings of the bypass valve opening and closing by the force from the stepper motor.

Controls

Controls are placed outside the unit for the prototype, but a curvet room in the top part of the indoor lid is reserved to controls.

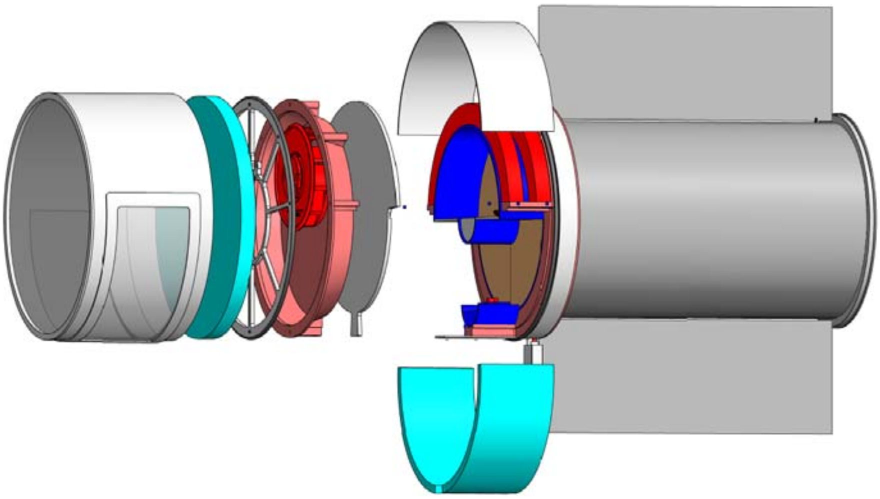


Figure 7 - An exploded view of the prototype. The room for controls are placed underneath the two red bars.

Construction

The tube around the heat exchanger is the main structure of the unit and we add all other parts to this. The slim pressure chamber and the fan for the exhaust air are contained in an extension of the tube on the heat exchangers external side. On the internal side of the heat exchanger, the end of the tube is closed by parts around the pressure chamber and the fan for the inlet air.

A weakness in the current design is in the interior lid, where 2 O-rings are placed to secure an air tight connection between the large cap (the grey piece furthest to the left on figure 7) and the rest of the interior lid. These O-rings have not been air-tight. Larger gaskets with another geometrical design should therefore be implemented.

Design of visual parts

As explained above, the internal lid needs to be rather large. At the same time large parts of the surface areas needs to be open for airflows through filters. In fact, the lid is a light, cylindrically formed cap, which covers the internal side of the unit. The entire front is a perforated aluminum plate which covers the M5 filter and allows the inlet air to blow slowly into the room. In the bottom half of the cap is another perforated area through which the exhaust air enters the G4 filters. The design is kept so simple as possible to be discreet.

On the façade we use a standard ventilation grill of aluminum, as seen on figure 8.

Figure 8 - Photos of the interior and exterior visual parts. The interior on the right and the exterior on the left.

Production

Production strategy

The aim was to construct a prototype that naturally conformed to the design requirements, but also relied mainly on fairly easily machined plastic parts.

In addition, to minimize leakage care was given to manufacture the parts so that interfaces were as tight and uniform as possible via perpendicular alignment, and that effective sealing was sterically possible by either glue, silicone or packing material.

Heat exchanger

The two layers of foil was 0.35 mm transparent Mylar A (a modified polyester with high strength and rigidity) in lengths of almost 9 m. Silicone bump-ons was added to function as spacer material according to the theoretical design pattern, and the foil was rolled up against a Ø63 mm centre tube and aligned to the foil length of “perfect” winding in the spiral (marked in advance, see figure 9).

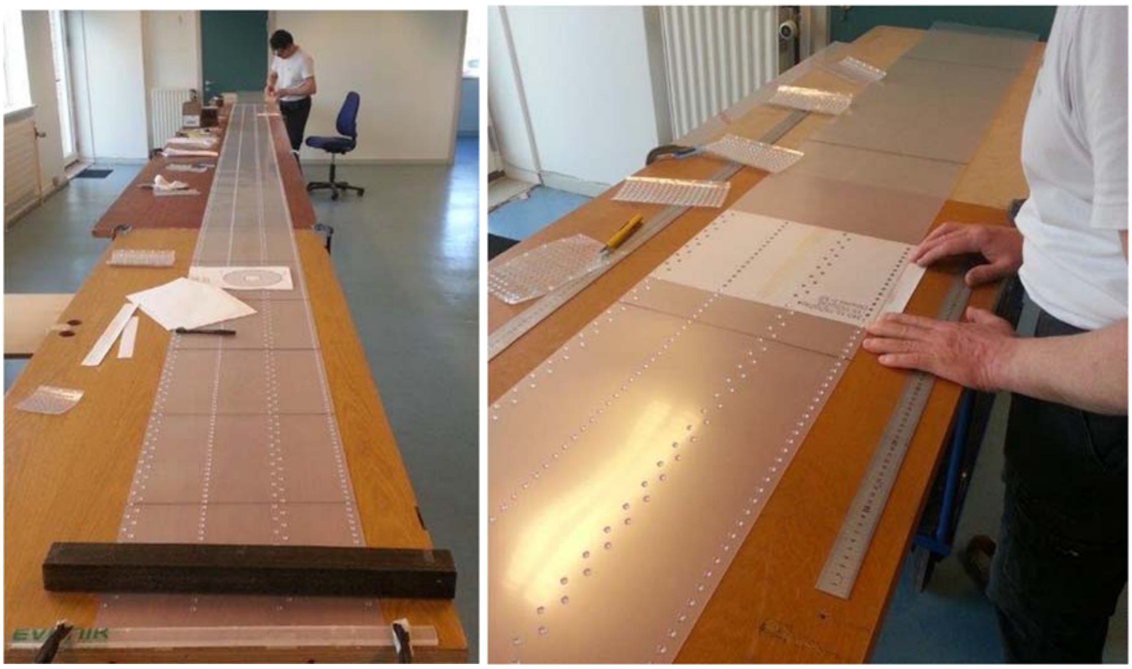


Figure 9 - *Adding spacers (bump-ons) and lining up spiral windings (black lines)*

After aligning the rolled up foils in a supporting grey PVC tube, every second layer was gap-filled with white acrylic silicone (see figure 10). After initial drying, another layer of silicone was added to fill up any visible air-bubbles (which would otherwise lead to internal leakage, possible with a pronounced effect). The outer layers were in the end of the process sealed against the PVC tube, thereby constituting “the heat exchanger” as a whole.

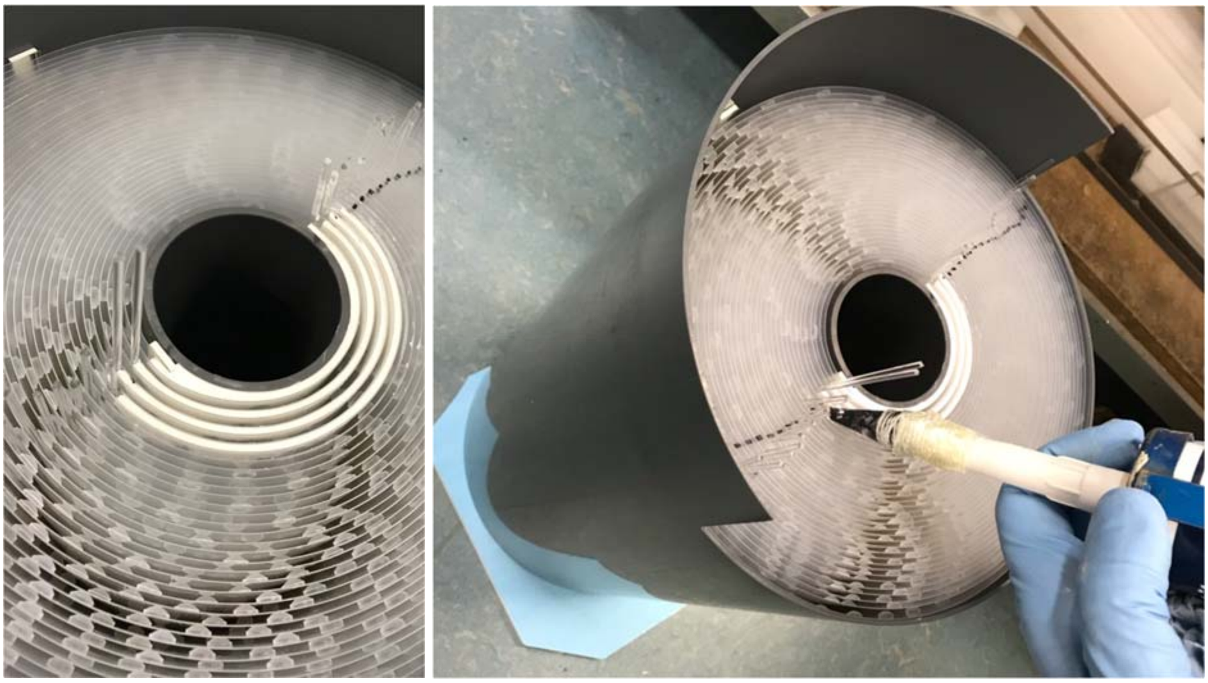


Figure 10 - Sealing the rolled up foil layers in the making of the functional heat exchanger. Clear rods acted to separate foils during sealing.

Tubes

A standard Ø250 mm PVC tube cut to length was used as the tube in which the unit is to be placed, i.e. serving as the interface between wall and unit. The same tube dimension was used to make the Ø240 mm outer tube of the heat exchanger by slicing up the tube, removing a strip of material and sealing the tube again.

The indoor lid of the unit was constructed from a custom-made Ø316 mm tube, since no tube dimensions with a sufficient tolerance was available in that size. Although more laborious to produce, this also gave the opportunity for choosing the colour, in this case we opted for white (see figure 11).

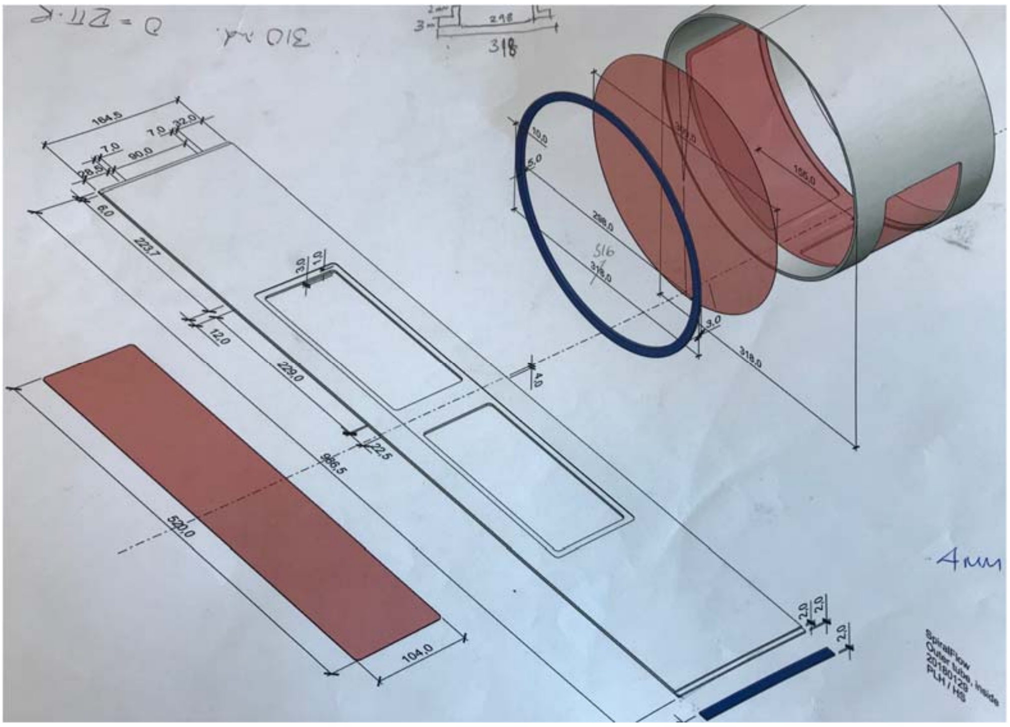


Figure 11 - Work drawing for making the tube part of the inner lid in 2 mm white PVC.

Milled parts

The vast majority of parts for the unit were 2D or 3D CNC milled in standard transparent or grey PVC in thickness of 5-50 mm (see picture S). In case where distances between parts would be crucial for keeping leakage at a minimum, the surface of the sheet would be milled down a defined thickness, thereby also ensuring perpendicular placement overall.

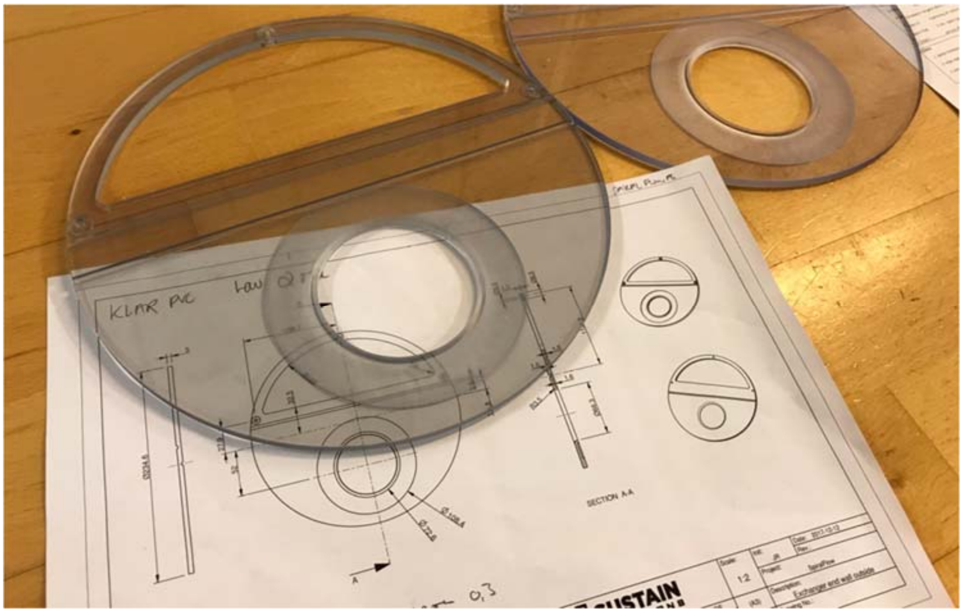


Figure 12 - Example of 2D CNC milled part; the item is used next to the outer lid, contributing to separation of air streams.

Vacuum formed parts

The two horizontal plates lined up against each side of the heat exchanger ensured separation of the air streams. Therefore, the integrity of this area was of outmost importance. To have the fewest number of sealing- and hence potential leakage points – we vacuum formed these two plates from a sheet of transparent PVC (see Picture T). This was an iterative process, adjusting the shape of form to represent the precise geometry of the inner Ø63 mm tube.

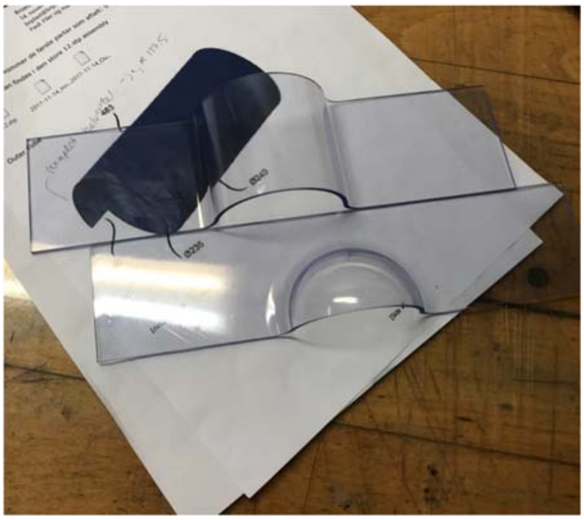


Figure 13 - Pictures of the vacuum formed parts.

3D printed part

The only part that could not be readily machined was the bypass unit that is centred in the inner tube of the heat exchanger. The complex geometry and diminutive details in combination with the requirement of it being very tight during normal, closed mode, made us purchase a 3D print of this.

Future production

Care was given during the design phase to make most of the parts of the Spiral flow unit easy to produce, either through machining or in the future even injection moulding. We succeeded in making the majority of parts as simple as possible, while the heat exchanger in itself would have to be produced more cost-efficient.

The material characteristics of the Mylar A foils makes it suitable for forming, and the very laborious and expensive method of using adhesive silicone bump-ons for spacer material, might be substituted by imprinting the “bumps” in the foil mechanically or via thermoforming. Mylar A foils has successfully been imprinted for other applications in a continuous process termed “rotary puncturing”, in which the foils in run between two rollers with a motif under pressure. Given that this is a continuous process, it might be a very fast and reliable production method.

Also the sealing of the foil layers was made by hand during the prototype making. This was a timely part, and the sealing should be made in some automated manner.

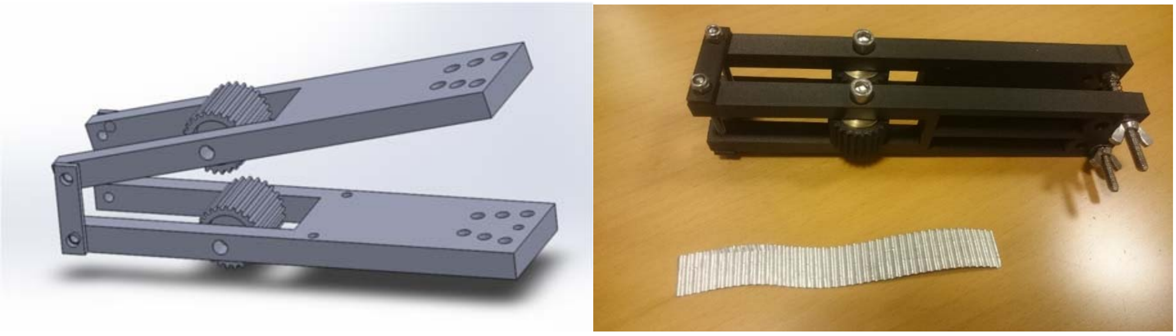


Figure 14 - 3D-printed corrugator tool. Used for corrugation of plastic sheets in the heat exchanger.

The project team attempted to create their own corrugated spacer with various thicknesses which could replace the bump-ons in the heat exchanger.

Existing corrugated materials already exists (mostly aluminium and plastic), but a thorough investigation has revealed that none of the already existing versions has as small dimensions as is needed in the heat exchanger.

Therefore, we designed and 3D-printed a frame and three sizes of corrugated wheels. We achieved the intended pattern with the wheels, but the grooves were not deep enough. We tried with several thicknesses of aluminium sheets, which a company supplied as samples. All of these were too thick, so we used a thinner, more malleable sheet from our workshop. Unfortunately, even the thickest corrugator could not create enough thickness in a spacer. The thickest we could achieve was 1.6mm as seen on figure 14.

The idea of producing our own corrugated spacer for the prototype was therefore discarded. It could however potentially work in an industrial setup if the correct material and methods are applied. This will need more investigation.

Predicted performance

Filters

The project team proposed to use a coarse filter upstream of the heat exchanger to protect it and a finer filter downstream of the heat exchanger to filter air for the occupant. Using a two-stage filter increases the time between filter changes since each filter removes its intended range of particles. The pre-filter will be coarse enough that it rarely needs to be changed. The occupant will need to replace the finer filter, but it will be closer to the occupant and thus easier to change. See the schematic below for clarification.

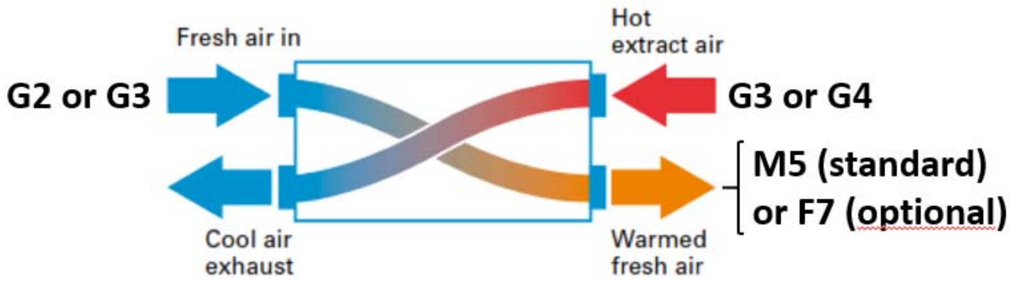


Figure 15 - Overview of filter placements in Spiralflow.

Perhaps most importantly, this design allowed us to increase the surface area of the supply filter by moving it out of the narrow cylindrical part of the unit. This gives less resistance to flow, which results in lower fan power and less fan noise. This is important when considering offering a fine F7 filter for removing pollen and other small particles since these filters give much higher resistances. Additionally, the larger filter area would require less frequent filter changes. A doubling in area results in a doubling of time between filter changes.

The pressure loss is important for selection of the fan. The design required a duty point (i.e. flowrate and pressure) in the optimal range of the fan to ensure efficiency and minimal noise. The geometry of the design determined our fan option, which in turn determined the maximum pressure loss for a given airflow. At 15 L/s, the predicted pressure drop of the exhaust filter was only 8 Pa. On the supply side, predicted pressure loss of the G2 pre-filter and M5 post-filter were roughly 8 Pa and 16 Pa, respectively.

The project decided to use a perforated steel sheet to diffuse the supply air into the room. All of the considered materials had a void ration greater than 45%. Calculations predicted a negligible effect on pressure loss from these sheets.

Heat exchanger

The heat exchanger was a focus for research and development. The spacer material added some pressure loss inside the heat exchanger and this affected airflows in different ways. There were thoughts to use the spacer material to achieve better airflow by forcing air to otherwise stagnant corners. The spacers could include corrugated material, stick-on bumpons or flow-straightening sticks. To study these effects, we simulated several configurations of spacers inside the heat exchanger using FEM simulations in Comsol. The figures below show the air velocity in a sample of these, including a clear layer with no spacers, a layer with continuous corrugation through the middle third of the heat exchanger, two 15 mm strips of corrugated spacer material at one-third intervals, and flow straightening sticks at various distances through the middle of the heat exchanger.

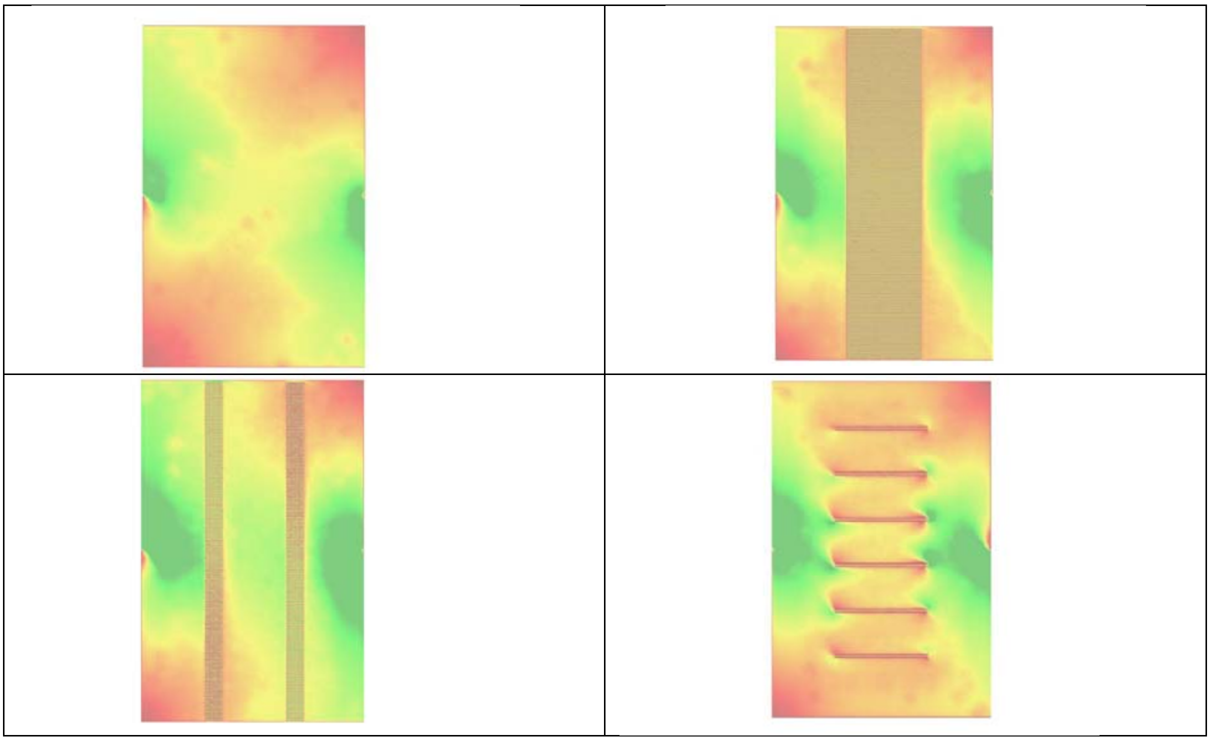


Figure 16. Velocity maps of airflow in a single layer of the heat exchanger (0-1 m/s) for different spacer configurations. The air entered the layer from the top left and exited through the bottom right.

The image below show the air velocities with circular bumpons at one-third intervals down the middle of the heat exchanger and distributed across the inlet and outlet of each layer. This is geometrically similar to the prototype. We hoped that the bumpons used for stability of the heat exchanger would force air to the corners, but this did not occur in simulations, or at least not significantly. However, the image to the right includes velocity arrows that show mostly longitudinal airflow through the middle portion of the heat exchanger, which indicates a counter-current velocity field for alternate layers. Which was what we aimed for.

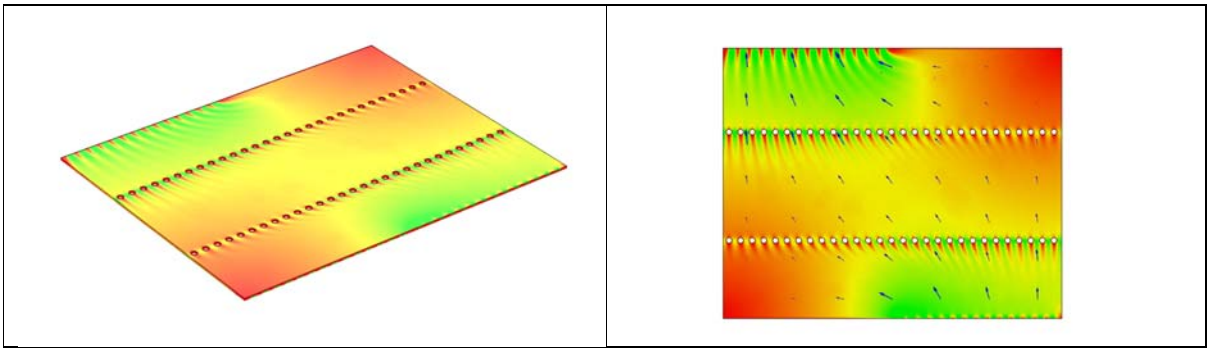


Figure 17. Velocity maps of airflow in a single layer of the heat exchanger (0-1 m/s) with a configuration of bumpons similar to the prototype.

The simulations also predicted the pressure loss to be 11 Pa for a flow rate of 6 L/s. This was a conservative estimate since it did not account for the contraction entering the heat exchanger or the expansion leaving it. During experiments, we measured the pressure loss for multiple flows and interpolated a value of 14 Pa for a flowrate of 6 L/s, so the predicted and measured values compared well.

Fans

We used an iterative process to select the fans and minimise the cumulative pressure losses through the unit. This ensured that the fans would operate in their optimal range of duty points (i.e. airflows and resistances) to maximise efficiency and stability while minimising fan noise. After predicting the pressure loss of all components, we expected to use a centrifugal fan. A forward-curved centrifugal works well against stronger counter-pressures, but it requires a scroll housing, which we could not fit into our design while maintaining enough surface area for the supply filter. We learned from previous experiences that we should aim to use the whole scroll to maximise efficiency or else select a fan without one. We decided to use a backward-curved centrifugal fan since it does not need a scroll. We set a restriction on pressure loss of 80 Pa for 15 L/s since this fit into the optimal range of both the fans we considered, as seen in the figure below. On the supply side, we chose the REF125 because it was available with a drive that minimised noise. The figures below show the optimal range of each fan and the duty point of 15 L/s and 80 Pa resistance. As can be seen from the curves, both fans offer excellent stability for changing resistances, which is important due to the effects of wind on the façade.

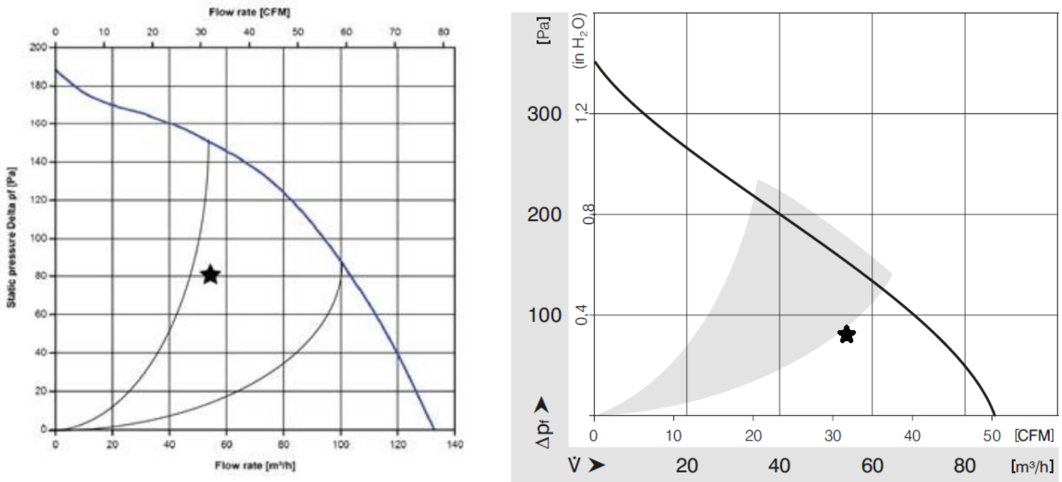


Figure 18 - Fan curves for the fans used in Spiralflow. Air flow is on the x-axis and pressure in on the y-axis.

Bypass

The first proposal for the bypass valve through the centre tube of the heat exchanger was a plug that opened 1 cm using a linear actuator. The figure below shows a 2D simulation that predicted the airflow inside the centre tube. The vertical line on the left represents the axis of rotation. The concern with this solution was that the gap would provide too much pressure loss. The figure shows the velocity field for a pressure difference of 48 Pa from end to end, which is the pressure loss of the heat exchanger at roughly 16 L/s. The average velocity exiting the bypass was 4.45 m/s, which yields a bypass airflow of 14 L/s. At a fully open position, we would expect the vast majority of flow to enter the bypass, so this gap for flow was insufficient. The pressure loss of this solution was likely too high since we dimensioned the tube to have less than one-tenth pressure loss of the heat exchanger. We could not open the plug any further with the considered linear actuators, so we turned to another solution. The rotating damper did not present any issues with respect to pressure loss.

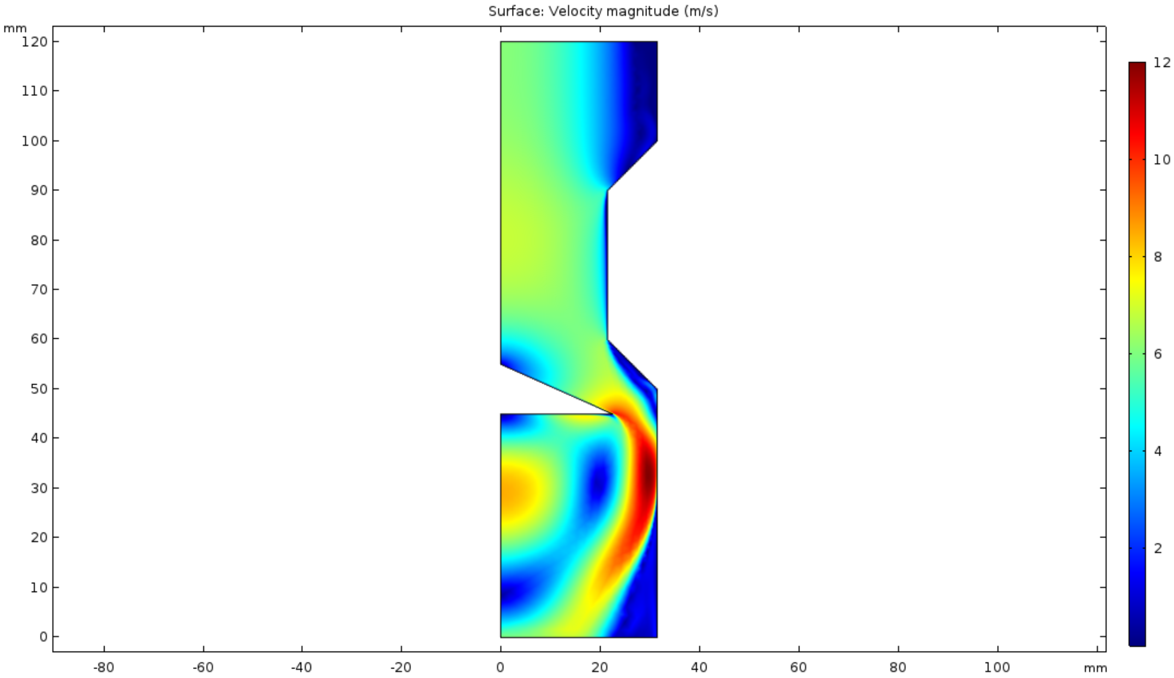


Figure 19 - Simulation of air flow through the bypass valve and in the centre tube.

Experimental setups and measurements

Heat recovery, air flow and power consumption

We measured the temperature efficiency of the heat exchanger in multiple ways.

We tested the heat exchanger in a guarded hot box, which offers a customisable insulated wall between two temperature-controlled chambers. We designed and built a test apparatus to the heat exchanger to control the supply and exhaust airflows, as seen in the figure below, where the green semi-circle represents a centrifugal fan.

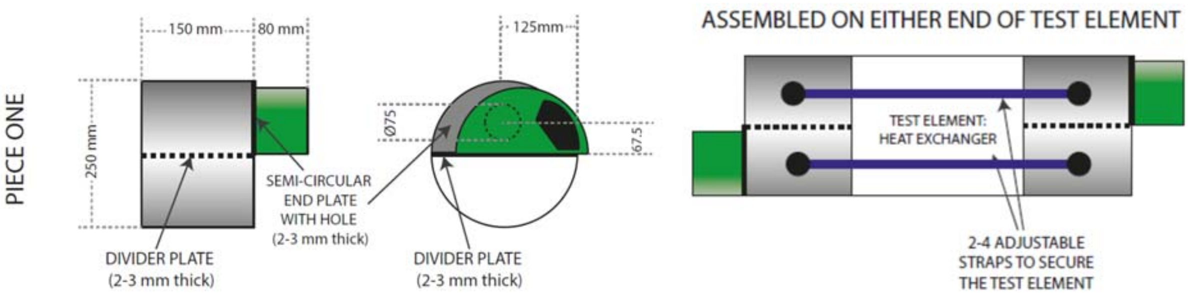


Figure 20 - Overview of test setup for heat exchanging effeciency.

After installing the device and setting the temperature for both chambers, we used four thermocouples at each inlet and outlet for the supply and exhaust airflows. We then calculated the supply and temperature efficiencies based one the following simple equations.

$$\eta_{supply} = \frac{(T_{supply} - T_{outdoor})}{(T_{indoor} - T_{outdoor})}\ ;\ \eta_{exhuast} = \frac{(T_{indoor} - T_{exhaust})}{(T_{indoor} - T_{outdoor})}$$

While the previously shown simulations indicated adequate heat transfer in all layers, we wanted to be certain that potential cross-flow in the outer layers did not decrease overall temperature efficiency, so we tested the prototype of the heat exchanger before and after blocking its outer two layers. As seen in the figure below, the fan signals provided similar temperature efficiencies before and after blocking these layers, but the airflows decreased by 10% to 20%. This indicated that the outer layers helped overall performance, so we should not decrease the diameter. This was an important design consideration because we had trouble fitting these layers into the standard size tube. This motivated an effort to make it work, which we did.

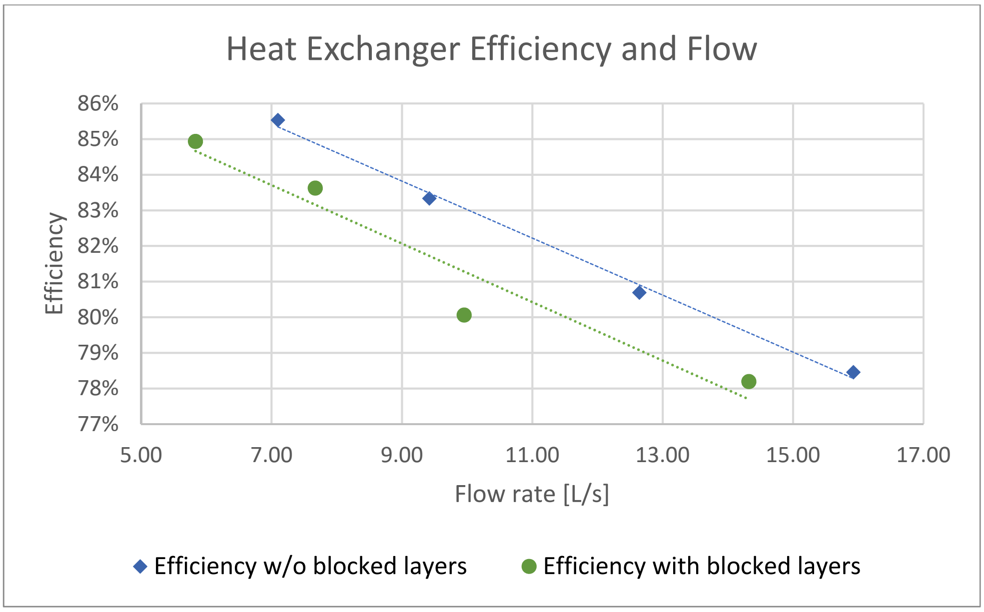


Figure 21 - Heat exchanging efficiency with open and closed outer layers of the heat exchanger to determine the outer layers impact on performance.

As seen in the figure below, the blocked outer layers resulted in much higher pressure-losses for each airflow. The project team targeted a total pressure loss for the whole unit of 70 Pa for 15 L/s, so including the outer two layers helped to meet this aim.

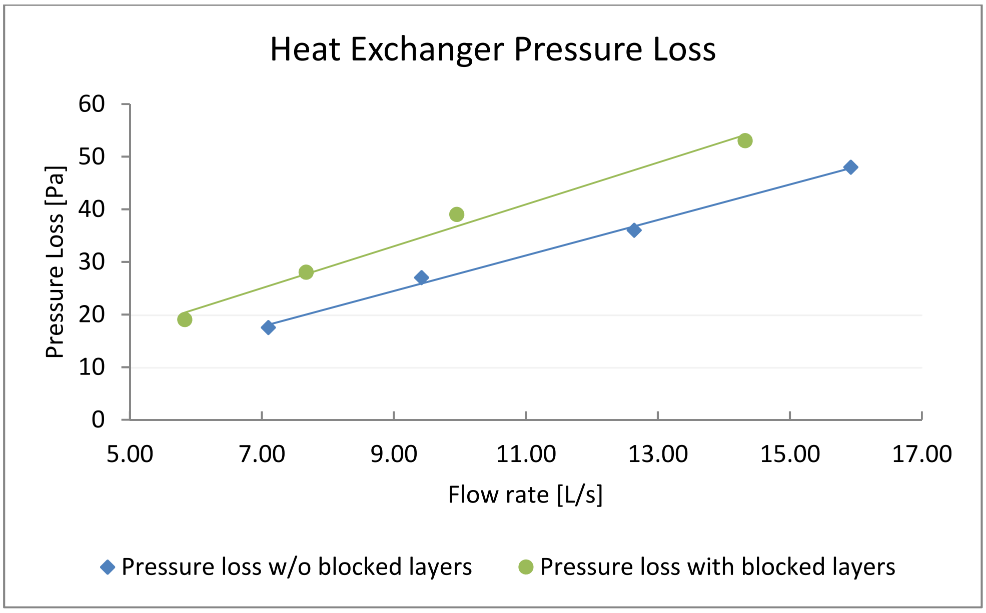


Figure 22 - Pressure loss through the heat exchanger. With open and closed outer layers of the heat exchanger to determine the outer layers impact on performance.

It is worth noting that during both efficiency tests, we measured internal leakage to be sure that none of the seals had shifted and invalidated the results. The leakage was so low that it was unmeasurable.

Once we constructed the prototype, we measured the airflow rates for each fan signal. We used these to formulate balanced flowrates and measured the corresponding temperature efficiencies. We placed the full prototype into an insulated panel in the wall of a test house at DTU. We drilled holes in the perforated diffuser to gain access for the thermocouples.

Figure 23 - Spiralflow unit attached with thermocouples to measure its heat exchanging efficiency.

Initially, we measured temperature efficiencies that were significantly unbalanced despite measuring minimal internal leakage. The heat exchanger is symmetrical on supply and exhaust, so this was an unexpected result. Since the prototype used two different fans, and the exhaust airflow measurements were prone to error, we decided to adjust the exhaust airflows to achieve balanced temperature efficiencies. This biased the results, so the results in the figure below are preliminary, but they satisfied our requirements for the prototype stage. The results imply greater than 85% temperature efficiency for airflows of 10 L/s. This surpassed the performance of the heat exchanger we had measured previously, which may be due to its improved construction for the prototype.

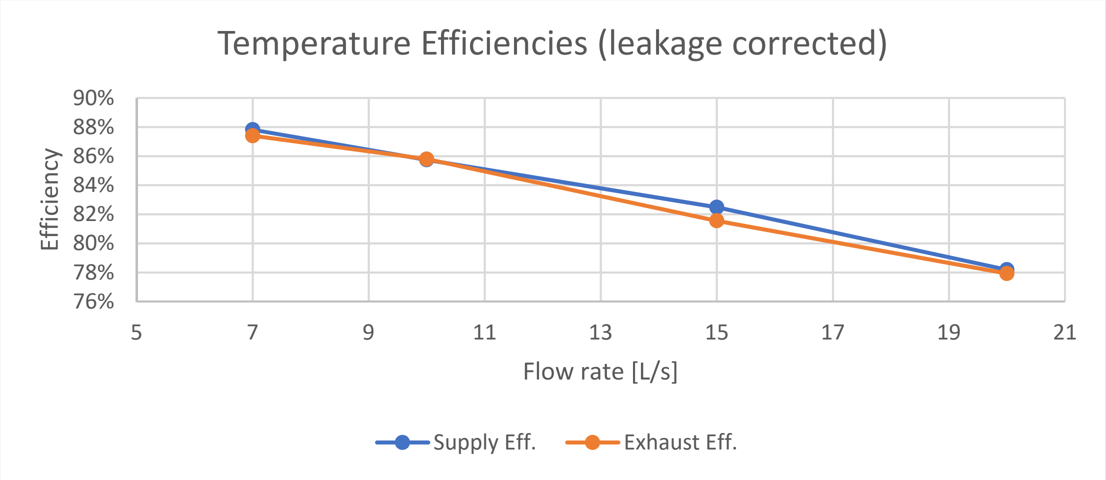


Figure 24 - Heat exchanging efficiencies of Spiralflow after a correction for leakage has been applied.

Air flow and Power consumption

We used Lindab FTMU flow meters to measure the airflow rates on the supply and exhaust air. The FTMU is an ultrasonic device and offers very little pressure loss. We added 5 meters of Ø100 mm duct before the flow meter and 1 meter after. We measured the supply airflow on the intake side using an attachment that we designed and 3D printed due to the irregular geometry of the prototype. The printed attachment is shown below.

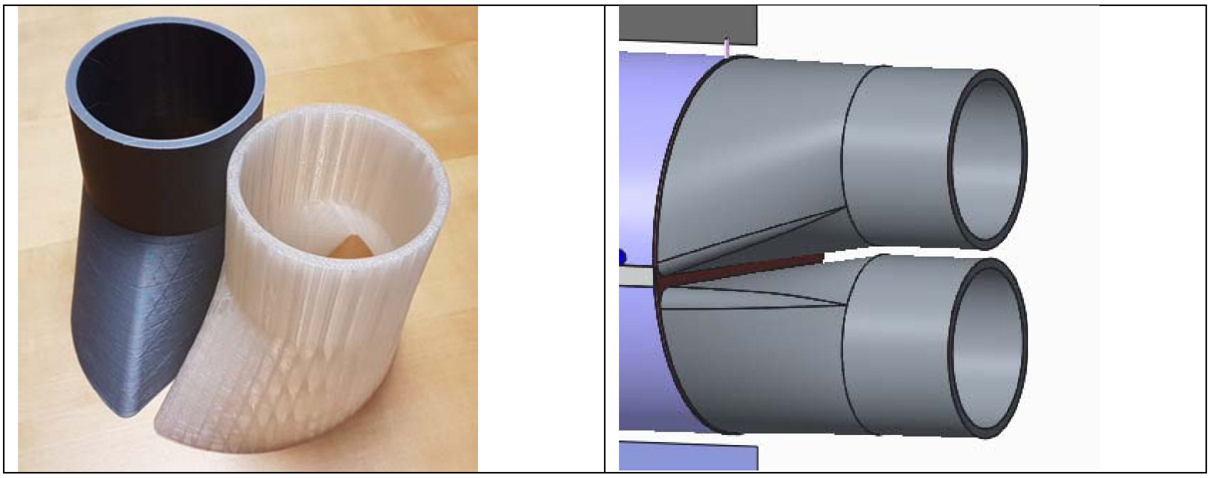


Figure 25 - 3D-printed attachments used to measure air flow through the unit.

The following figure shows the measured flow rates on the supply side as well as the specific fan power, which is the required fan energy to move one cubic meter of air (J/m^3). The 2020 building regulations place a limit on specific fan power (SFP) of 800 J/m^3 for both fans if a ventilation unit serves a single dwelling, so the figure shows a limit of 400 J/m^3, corresponding to a single fan.

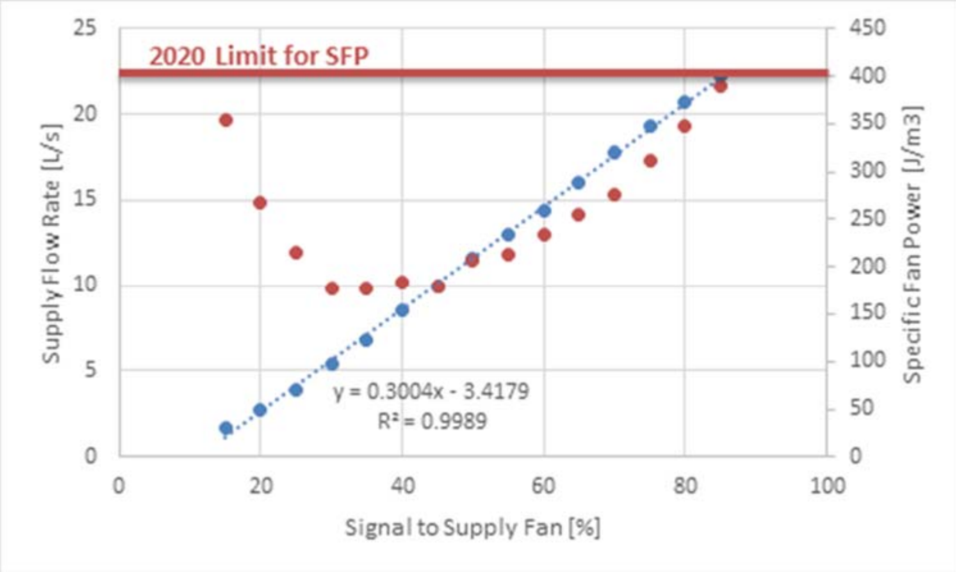


Figure 26 - Air flow and Specific Fan Power (SFP) for the interior fan. The blue curve is air flow and corresponds to the left y-axis. The red curve is the SFP and corresponds to the right y-axis. The limit for the Danish building regulation 2020 is set to 400 J/m3 for one fan.

The following plot shows results from the same measurements on the exhaust side. We used the same 5-meter length of Ø100 mm duct ahead of the flow meter, but the flow meter gave varying results as we tried rotating it, which it should not. This call into question the claimed accuracy of the flow meter when used as direction, as it claimed better than 5% error. As a result, these measurements only provide a rough indication of flow rates and fan powers (i.e. SFP). Again, the preliminary results show very low fan power for all flow rates.

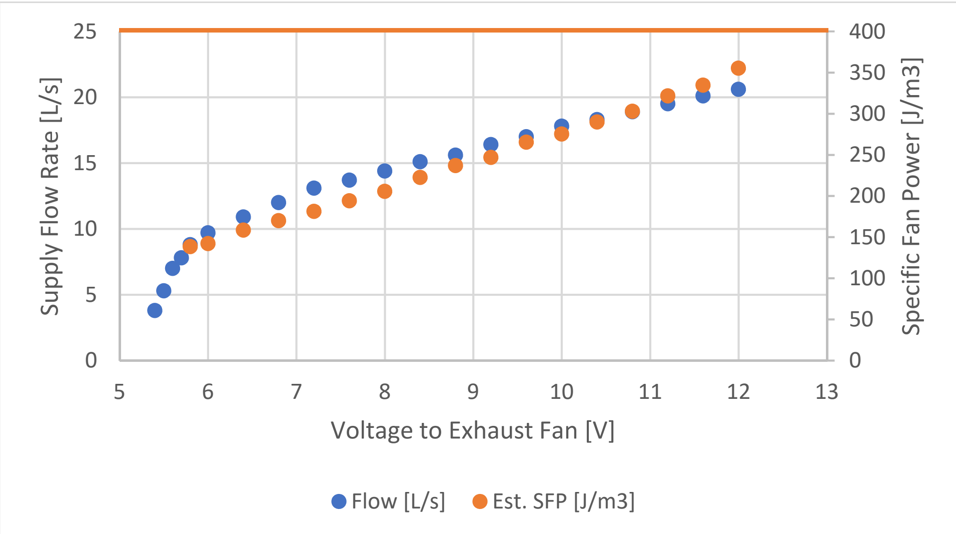


Figure 27 - Air flow and Specific Fan Power (SFP) for the exterior fan. The blue curve is air flow and corresponds to the left y-axis. The red curve is the SFP and corresponds to the right y-axis. The limit for the Danish building regulation 2020 is set to 400 J/m3 for one fan.

While the exhaust flowrates carry significant uncertainly, it is worthwhile to show the total SFP for both fans, as the results indicate a very successful design of a low-pressure system. The figure below shows an SFP below 800 J/m^3 up to 22 L/s. This may improve further as we optimise the location and dimension of the fans in future prototypes.

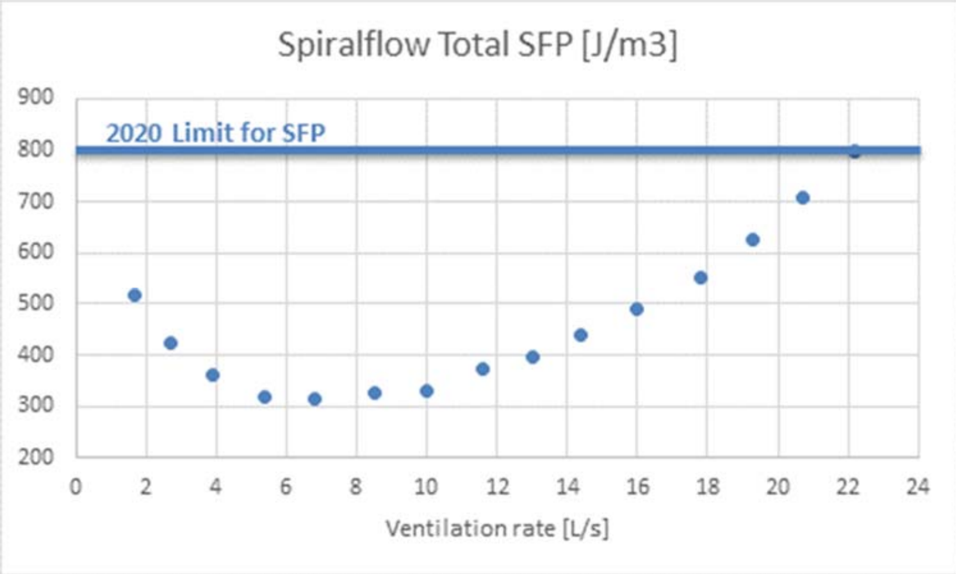


Figure 28 - The total specific fan power for both fans. The graph shows that the unit is below requirements for the 2020 regulations for all flow rates.

Due to the aforementioned uncertainty of measurements on the exhaust air, we measure the flowrates again with tracer gas for two sets of signals for supply and exhaust. These measurements were not continuous, so they also carry potential for error, but they provide another helpful indication of airflow. We dosed 2 L/min of Dinitrogen Oxide as a tracer gas using a rotameter and measured tracer gas concentrations with an Innova Analyser and Multiplexer. The following table shows the estimates of airflows based on tracer gas measurement. The table also shows the measured values using the flow meter for comparison.

	Concentration [ppm]	Airflow – Tracer [L/s]	Certainty [qualitative]	Airflow – Meter [L/s]
Exhaust 7.0V	2789	12.0	Reasonable	12.6
Supply 50%	2656	12.6	Poor	11.6
Exhaust 9.4V	2128	15.7	Poor	16.7
Supply 67%	1997	16.7	Poor	16.7

Table 1 - Air flow results based on tracer gas measurements compared to air flow measurements performed by air flow meters.

Leakage

We used tracer gas to measure the internal and external leakage and mixing. We tested according the EN Standard 13141-8, which dictates the following four measurements. The internal leakage on exhaust air (from exhaust to supply) was only 1.50%, while the internal leakage on outdoor air (from supply to exhaust) was 3%. Both of these are within the limits of category 1. This validated earlier measurements of pressure leakage based on flow rates and pressure differences. Combined, these measurements showed adequate sealing inside the heat exchanger.

As per the standard, we measured the internal leakage on exhaust air plus outdoor short-circuiting/mixing. We performed a measurement then added extra sealant before performing another. The results were 31% and 26%, respectively. We also measured the internal leakage on outdoor air plus indoor short-circuiting/mixing. We took three separate measurements, which ranged from 18%-22%. The indoor and outdoor short-circuiting demonstrated that we should focus on creating more separation between inlets and outlets in future prototypes to reduce these values.

Acoustics

Sound Transmission Loss

Tests of transmission loss and sound power were conducted in the laboratory test facilities at DTU. The tests rooms used were rooms 004 and 003. These are reverberation rooms of 230 m^2 and 215 m^2 respectively, with a 10 m^2 niche opening between them. The rooms are equipped with diffusers of varying sizes on walls and ceilings, ensuring a diffuse sound field.

Measurement setup

The test setup has two corner loudspeakers in the source room (004) generating the sound field. In both source room (004) and receiving room (003) the energy averaged sound pressure is recorded in 1/3rd octave bands, from 50 Hz to 5 kHz, with a microphone on a rotating boom. The boom radius is about 1.2 m and rotates with a revolution time of 16 s. The distance from the microphone is, at any time, at least 0.5 m from a surface and 1.0@m from the source. The averaging time is four revolutions, 64 s. The measurement is repeated with each source position. The background noise and reverberation time is recorded in the receiving room only.

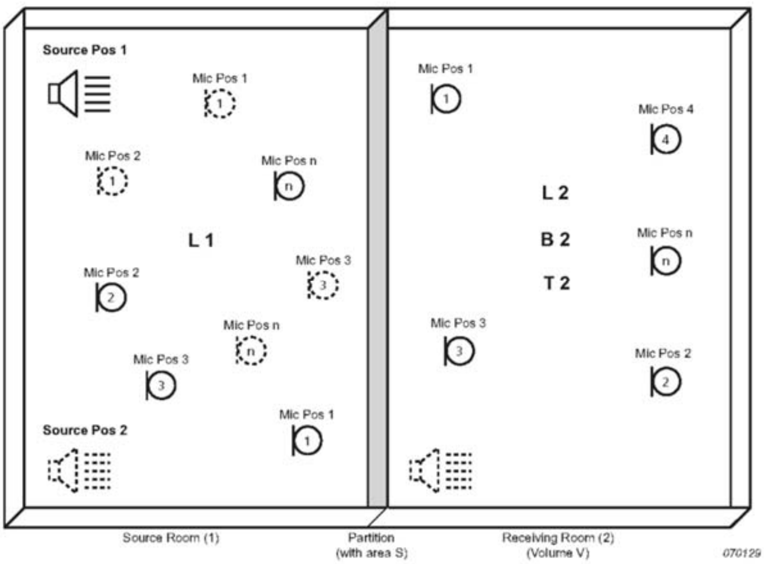


Figure 29 - Typical setup for sound insulation measurements.

For the reverberation time, the microphone boom is slowed down to a revolution time of 64 s, and the sound decay is recorded with interrupted noise in 6 arbitrary positions on the microphone path. The sound source used is a corner loudspeaker in the same room. Given the long reverberation time and the low background noise level of the laboratory, T30 is used for the evaluation.

All measurements and measurement procedures have been made in accordance with DS/ISO 10140-2:2010.

Mounting in the laboratory

A lightweight partition wall was built in the niche opening, where the ventilation unit was mounted during tests.

The airborne sound insulation was measured for the partition wall alone. The Sound reduction index of the partition wall is R*w* = 64 dB, with spectral corrections C and Ctr of -1 and -7. The Ventilation unit was mounted decentrally in a circular opening, Ø = 253 mm. With at least 1.2 m to the niche boundaries. The unit was positioned in the cut-out, covering the hole at least 3 cm from the edges.

Airborne sound insulation

For a large building element, e.g. a partition wall or a door, the Sound Reduction index, *R*, is reported, whereas for small technical elements it is customary to report the element-normalised level difference $D_{n,e}$. Both parameters are measured using the same setup and based on the level difference between source and receiving room.

Sound reduction index, R

The sound reduction index is found for each 1/3rd octave bands as:

$$\mathrm{R} = L_1 - L_2 + 10 \log 10\left(\frac{S}{A}\right)$$

Where
L_1 is the energy averaged sound pressure in the source room, in decibels;
L_2 is the energy averaged sound pressure in the receiving room, in decibels;
A_0 is the area of the free test opening in which the element is installed, in square metres;
A is the equivalent absorption area in the receiving room, in square metres.

Element-normalised level difference, $D_{n,e}$

For smaller elements, it is useful to report a level difference of the element, rather than an area dependent sound reduction index. This makes a description of the element, which is not changing with mounting, but is simple to include in a calculation of a given construction. The element-normalised level difference is, very similarly to the sound reduction index, found as:

$$\mathrm{D}_{\mathrm{n,e}} = L_1 - L_2 + 10 \log 10\left(\frac{A_0}{A}\right),$$

where
L_1 is the energy averaged sound pressure in the source room, in decibels;
L_2 is the energy averaged sound pressure in the receiving room, in decibels;
A_0 is the reference absorption area in square metres (for the laboratory, A_0 = 10 m2);
A is the equivalent absorption area in the receiving room, in square metres.
Is is worth noting that for a test object of 10 m^2, $D_{n,e}$ will be equal to R.

Weighted single number value

On the basis of the calculations of either R or $D_{n,e}$, a single number weighted value is found. The evaluation is based on a reference curve, which is shifted in steps of 1 dB until the least favourable deviations between the curve and the measured values are as close to, but not more than 32 dB. The least favourable deviations are found where the measured parameter is below the reference curve. The reported weighted single number value is found as the value of the evaluation curve at 500 Hz. The evaluation method is described in detail in DS/ISO 717-1:2013.

The airborne sound insulation has been measured for the ventilation unit currently on the market, the BREATHE 55 and the Spiralflow working prototype in order to compare Spiralflow unit with already market available units.

BREATHE 55

The element-normalised level difference of the BREATHE 55 ventilation unit:

$$\mathrm{Dn,e,w\ (C;Ctr)} = 45\ (-1;-3)\ \mathrm{dB}$$

The spectral values and evaluation curve can be seen in Appendix A.

Spiralflow

The element-normalised level difference of the Spiral Flow ventilation unit prototype is found to be:

$$\mathrm{Dn, e, w}\ (\mathrm{C; Ctr})\ =\ 40\ (-1; -3)\ \mathrm{dB}$$

The spectral values and evaluation curve can be seen in Appendix B.

Sound Power

Measurements of sound power has been carried out for the indoor and the outdoor emission of the ventilation unit. With the mounting in the reverberation room, the measurement follows the procedure of ISO-3741:2010. All values of sound power level are reported as A-weighted values.

Measurement Setup

For the sound power measurements, the room averaged sound pressure level has to be determined with at least 6 dB difference to the background noise of the setup, preferably a 10 dB difference. This is usually not an issue with high power sources, such as vacuum cleaners, compressors, or even washing machines. Whereas a ventilation unit with a regulatory limit of LAeq = 30 dB(A), measured in a furnished room, will likely have a very small sound power, and it is important to ensure a very low background noise level.

Sound power level

The sound power of the source is calculated from the room averaged sound pressure level by accounting for the reverberant conditions in the room, assuming a diffuse sound field, and the room geometry, given by surface area and volume. The equation used is:

$$L_W\ =\ \overline{L_p}\ +\ \left\{10\log\frac{A}{A_0} + 4.34\,\frac{A}{S}\ +\ 10\log\ \left(1\ +\ \frac{S\ \cdot\ c}{8\ \cdot\ V\ \cdot\ f}\right) -\ 6\right\}$$

where

L_W is the sound power level of the sound source under test (dB);
L_p is the average sound pressure level in the room (dB);
A is the equivalent sound absorption area of the room (m^2);
A_0 = 1 m^2;
S is the total surface area of the reverberation room (m^2);
V is the volume of the room (m^3);
f is the midband frequency of measurement (Hz);
c is the speed of sound

Note that due to the very stable temperature conditions of the laboratory, the a-correction for the temperature of the room has not been applied. All measurements where conducted at an air temperature of 20±0.5^{o}C. The total sound power is found as ten times the logarithm of the sum of the A-weighted 1/3rd octave band sound power levels.

$$L_{WA,Tot}\ =\ 10\log\left(\sum\ L_{WA}\right)$$

The sound power for the SpiralFlow prototype has been found under the following conditions:

- Supply fan only
- Exhaust fan only
- Both fans running

Each fan combination have been measured with four different air flows, which corresponds to the fan settings listed in table 2.

Air Flow	Supply fan	Exhaust fan
5 L/s	28 %	5.6 V
10 L/s	45 %	6.7 V
15 L/s	61 %	9.7 V
20 L/s	78 %	14.6 V

Table 2 - Settings for the fans in the SpiralFlow unit during test

Indoor sound power

Sound power has been calculated for air flow each setting, and measurements are grouped in column diagrams for each fan as well as the combination of the two, showing the spectral sound power, as well as the total sound power for all air flows. Sound power is presented for the interior in Appendix C.

LAeq values of ventilation unit

Since the setup of the investigative measurement is very similar to the insitu measurement ass listed in SBi-217, the measured *LAeq* values has been presented as single number values corrected for the longer reverberation time in the laboratory.

Air flow	Supply fan	Exhaust fan	Both fans	unit
5 L/s	31	34	36	dB(A)
10 L/s	42	45	48	dB(A)
15 L/s	51	51	54	dB(A)
20 L/s	58	54	60	dB(A)

Table 3 - Indoor LAeq *of the SpiralFlow prototype unit*

Air flow	Supply fan	Exhaust fan	Both fans	unit
5 L/s	29	40	41	dB(A)
10 L/s	38	49	50	dB(A)
15 L/s	44	57	58	dB(A)
20 L/s	50	64	65	dB(A)

Table 4 - Outdoor LAeq of the SpiralFlow prototype unit

Market analysis

In order to estimate the Danish market for the spiralflow unit has statistical data from Dansk Statistik been used. There has been developed an excel sheet which based on the wall thicknesses the unit can adapt to can estimate the potential market of kitchen and bathrooms in Denmark.

The focus has been on bathrooms, toilets, toilet+bathroom and kitchens. The main focus has been on multi-storey buildings, but the analysis also contains estimates for single family houses and cluster houses.

The approach to the analysis has been as follows:

- Determination of building typologies: A series of building typologies has been determined. This has taken its basis from ”Danish building typologies” by SBi, 2012. For each typology is a series of characteristics e.g. wall thicknesses.
- Determination of the building mass: The size of the total Danish building mass has been determined from statistical data from Danmarks Statistik. It has been divided into 3 main categories which each is divided into 9 sub-categories.
- Determination of wet rooms: From statistical data from Danmarks Statistik has the distribution of the number of kitchens, toilets and bathrooms been estimated.
- Determination of potential wet rooms: The total amount of wet rooms has been narrowed down by the demand that the wet room has to be facing the façade. This is due to the Spiralflow unit which only can be installed in the façade. The amount of wet rooms facing the façade has been estimated for each typology for multi-storey buildings by investigating roughly 200 floor plans. This has been done by investigating buildings for sale by www.boligsiden.dk. The data has been investigated from the end of May until mid-June 2017.

The model for the market analysis has been constructed so various parameters can be changed in order to investigate a more specific market with buildings with specific demands or that the specifications from the Spiralflow unit changes or new data from the building typologies has emerged.

Data og sources

- Danmarks Statistik
  - Table: BOL102, year 2016 (total number of housing)
  - Numbers from BBR, year 1981-2010 (the development has been projected with the average yearly change until 2016)
- Investigation of floor plans (www.boligsiden.dk)
- ”Danish building typologies” by SBi, 2012

An example of how the model works can be seen in D.

Results

By the minimum wall thickness of 390mm for the current prototype it is estimated that the potential Danish market of wet rooms is 700.000.

If the unit could be developed further so the minimum wall thickness could be 50mm less (340mm) the potential market would grow to 1.675.000.

Estimated sales

Based on the market analysis there has been developed a model to estimate the sales of the Spiralflow unit on the Danish market. There has been several assumptions and scenarios made in order to do so.

General assumptions:

- Project running in 7 years
- Development costs in 2018 and 2019
- Product ready to sell in 2020 until 2024
- Discount rate: 4 %
- Average sales price: 600 EUR
- Production cost (<1.000 units): 400 EUR
- Production cost (>1.000 units): 350 EUR

Scenario 1 and 2 (Small scale)

Assumptions

In scenario 1 and 2 it's assumed that there are 0,5 employees working on the project. Further assumptions can be seen in table 5 and 6.

Scenario 1 (Small Scale)

Assumptions	2018	2019	2020	2021	2022	2023	2024
Units sold	-	-	25	50	100	200	400
Operating costs, DKK	250.000	250.000	500.000	500.000	500.000	500.000	500.000
Development costs, DKK	250.000	750.000	100.000	50.000	50.000	50.000	50.000

Table 5 - Overview of assumption used in scenario 1.

Scenario 2 (Small Scale)

Assumptions	2018	2019	2020	2021	2022	2023	2024
Units sold	-	-	50	100	200	400	800
Operating costs, DKK	250.000	250.000	500.000	500.000	500.000	500.000	500.000
Development costs, DKK	250.000	750.000	100.000	50.000	50.000	50.000	50.000

Table 6 - Overview of assumption used in scenario 2.

Scenario 3 and 4 (Large scale)

In scenario 3 and 4 it's assumed that there are 4 employees working on the project. In the market research, it was found that the market contains 700.000 wet rooms. Assuming that the households only need to be renovated every 100 years (1%), only 50% of the households want to install ventilation and that there are 12 products (market share 1/12) on the market the units sold will be

$$292\ units \approx \frac{700.000 * 1\% * 50\%}{12}$$

In scenario 4 its assumed after the year 2021 that the market share is 1/6.

Further assumptions can be seen in table 7 and 8.

Scenario 3 (Large Scale)

Assumptions	2018	2019	2020	2021	2022	2023	2024
Units sold	-	-	292	292	292	292	292
Operating costs, DKK	1.000.000	1.000.000	4.000.000	4.000.000	4.000.000	4.000.000	4.000.000
Development costs, DKK	250.000	1.500.000	500.000	500.000	500.000	500.000	500.000

Table 7 - Overview of assumption used in scenario 3.

Scenario 4 (Large Scale)

Assumptions	2018	2019	2020	2021	2022	2023	2024
Units sold	-	-	292	292	583	583	583
Operating costs, DKK	1.000.000	1.000.000	4.000.000	4.000.000	4.000.000	4.000.000	4.000.000
Development costs, DKK	250.000	1.500.000	500.000	500.000	500.000	500.000	500.000

Table 8 - Overview of assumption used in scenario 4.

Results

From table 5 below we see that none of the scenarios has a profitable Net Present Value (NPV).

Scenario	NPV 2018, DKK	Internal rate of interest
1	-2.918.098	#NUM!
2	-2.032.979	-30%
3	-22.185.716	#NUM!
4	-20.845.545	#NUM!

Table 9 - Net Present Values (NPV) for the 4 different scenarios.

If the same assumptions are applied, but the market is thought not only to wet rooms, but also to dry rooms, which is thought to be twice as many as wet rooms. Tripling the market size. The analysis would be different as shown below. This reveals that only one of the scenarios has a positive NPV.

Scenario	NPV 2018, DKK	Internal rate of interest
1	-1.030.103	-9%
2	1.036.470	15%
3	-17.535.024	#NUM!
4	-13.514.510	#NUM!

Table 10 - NPV for the 4 different scenarios, with an inclusion of dry rooms in the market estimate, tripling the potential market size.

Dissemination

During the project 2 master students from DTU Design and Innovation has performed their master thesis on Spiralflow. The two students, Jakob Thomsen and Carina Lindahl looked into the problem of what to do with condensation from the unit and came up with several ideas regarding this based on interviews with stakeholders in the building industry. This resulted in the master thesis 'Product life design in the construction industry – The case of Spiraflow ventilation unit', which lead to the grade 12 for both Jakob and Carina.

Discussion

It was the ambition to install several prototypes in actual homes and perform interviews of the users in order to get feedback on the unit. This has shown to be too ambitious as it was not achieved. This was due to the production of the heat exchanger which took longer time than expected and required a lot of focus. This was necessary as prototypes would not be able to be constructed without a fully functioning heat exchanger.

Test results

The internal leakage of the unit is very low, this is due to the way the heat exchanger is constructed. But the short circuiting of air on both the interior and exterior side are very high. These short circuiting's will have a great influence on the overall ventilation effectiveness.

The short circuiting on the exterior side is estimated to be in the range of 26-31%. Which means that a quarter to a third of the air that the unit exhausts will be supplied back in. Since the grill is flush with the façade is the separation of the two air streams limited. That in combination with low velocities of air flow makes the high short circuiting. A deeper investigation of air flows and exterior grills could potentially bring the high short circuiting down. But the flush-with-the-façade design will most likely always experience short circuiting.

The short circuiting on the interior side is estimated to be around 18-22%. The large filter areas ensure a low pressure drop, but also a low velocity. It is believed that this high short circuiting is due to the low velocity. A potential solution could be to increase the pressure drop by decreasing the filter areas. By doing so the power consumption will increase and potentially will the noise level as well. Smaller filter areas could potentially mean that the size of the interior lid could be minimized.

Transmission noise dampening for the prototype is 40 dB(A), whereas for a unit on the market (Breathe55) the noise dampening is 45 dB(A). Since the Spiralflow unit is a prototype the installation of the unit has not yet been developed. During the installation sealants and other materials like this could help to improve the transmission noise dampening further. The unit is therefore thought to be competitive to similar products on the market.

Future market

By the market analysis and the estimated sales, the unit is currently not profitable. The potential market size is currently estimated only from households and the wet rooms within. The actual potential market might be larger as the unit isn't limited to wet rooms or households. It could therefore fit into dry rooms and e.g. office buildings. A market analysis taken this into account could potentially change the current non-profitable project into a profitable project.

Based on the current market analysis 3 potential developments are suggested:

1. Adapt to slimmer wall thicknesses
2. Lower development costs
3. Broaden the market by entering foreign markets

The first point relies on the heat exchanger and the efficiency of this. The minimum wall thickness is currently 390mm and the heat exchanger is currently 340mm in length. If the heat exchanger is developed further and the efficiency is increased, this could mean that the length could be decreased and thereby the length of the unit could be decreased. If this is not possible a change in the design of the inlet and outlet to the heat exchanger could be optimized in order to save space. This however has a limitation of the current length of the heat exchanger. A combination of the design and length of the heat exchanger is also possible.

The development costs are at this stage unknown and are there for an estimation. The development costs are currently estimated to be 1 mil. DKK, which is thought to be conservative. To get more knowledge on this the production of the heat exchanger needs to be investigated further.

From the estimated sales the NPV is only positive in one scenario. An assumption of an average sales price of €600 has been used. It should be investigated if the product potentially could bare a higher price in the market in order to improve the NPV.

The market analysis is a conservative estimate based on the current specifications of the prototype. As it is a first prototype with potential improvements, the market analysis in only thought to improve along with the improved specifications.

Another way to improve the NPV is to increase the market. If the market was broadened up to foreign countries the market share would increase. Entering a foreign market is however subjected to a cost. It is not known what the potential markets are neither are the cost for entering these markets. An investigation of this is recommended.

Conclusion

A functioning prototype has been constructed. The unit is flush with the exterior façade and is therefore very discrete.

The prototype has shown excellent results regarding, heat recovery, air flow and power consumption. Furthermore, it has shown good results regarding transmission noise and internal leakage. All of this achieved by the first prototype with clear indications of improvement. This is thought to be a fantastic result.

There is also room for improvement as the results for short circuiting and noise generation are not as good. Improvement ideas are already generated and ready to be implemented.

Future development is needed for the unit to go into production. Especially the roll-up of the heat exchanger needs clarifications.

The market analysis revealed that the unit should either be optimized so it can adapt to slimmer walls or the market size should be increased. A combination of the two could increase the market and potentially sales.

This project has proven that it is possible to construct a decentral ventilation unit with the new Spiralflow heat exchanger as a core-technology.

Perspective

Future development and production of Spiralflow

The heat exchanger is the key component of Spiralflow and needs to be further developed. This could potentially decrease the size and/or length of the heat exchanger, which is crucial for the market size. Downsizing the heat exchanger could potentially also mean that the interior lid could be minimized. This could also increase the market size.

The production of the heat exchanger is at this point unknown. An investigation of production techniques could reveal how it could be produced at a low cost, which has been the aim throughout the project. The process of imprinting the foils with bumps, so no bump-ons must be applied, will be crucial for the production cost.

A mechanical roll up of the heat exchanger could potentially increase the performance. If a technique of imprinting the foils is possible, this opens up the possibility of making an optimized pattern for the imprinting. Ensuring the sweet spot between pressure drop, uniform air flow distribution and heat recovery. This could potentially decrease the length of the heat exchanger, but potentially also the distances between the foils. Which means either more surface area and thereby higher efficiencies at the same space or the current efficiencies at a smaller space.

When possibly designing a new interior lid due to a more efficient heat exchanger. The connection in the interior lid by the large interior cap and the rest of the interior lid should be improved. Currently 2 O-rings are placed in the lid, but they have shown not to be effective regarding air-tightness. Other shaped gaskets should therefore be implemented. Here T-shaped gaskets are currently suggested.

By optimizing the fans and their placements can the performance of the unit be optimized both in regard to noise and power consumption. The 100mm fan is expected to be fully developed by Ebm-Papst during the fall of 2018.

Appendix A – Transmission noise of Breathe55

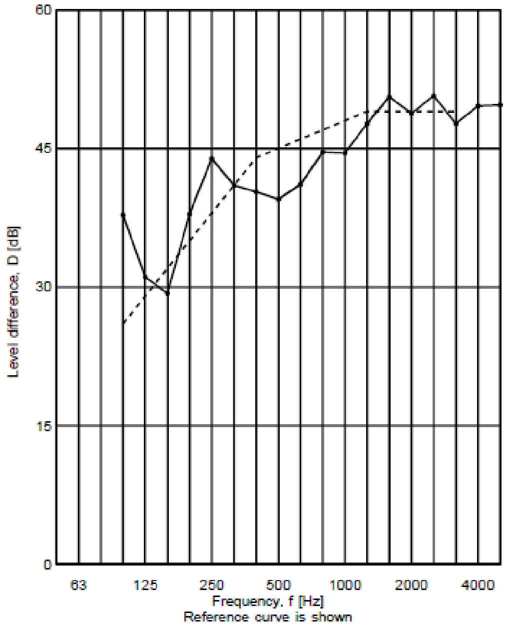
Data sheet 2

Laboratory measurement of airborne sound insulation of single element DS/EN ISO 10140-2:2010

Project: 56195
Date of test: 22 May 2018
Test location: DTU Building 355
Source room: 004
Receiving room: 003
Test object: BREATHE 55 ventilation unit

Partition area, S: 0.2 m²
Receiving room volume: 215 m³

Frequency f [Hz]	$D_{n,e}$ 1/3-octave [dB]
100	37.8
125	31.1
160	29.3
200	37.9
250	43.9
315	41.0
400	40.3
500	39.5
630	41.1
800	44.6
1000	44.5
1250	47.7
1600	50.6
2000	48.8
2500	50.7
3150	47.7
4000	49.6
5000	49.7



Weighted element-normalised level difference in accordance with DS/EN ISO 717-1:2013:

$D_{n,e,w}$ (C; C_{tr}) = 45 (-1;-3) dB

Based on laboratory measurements in accordance with DS/ISO 10140:2-2016

Udført af David Duhalde Rahbæk
Akustisk Teknologi, DTU

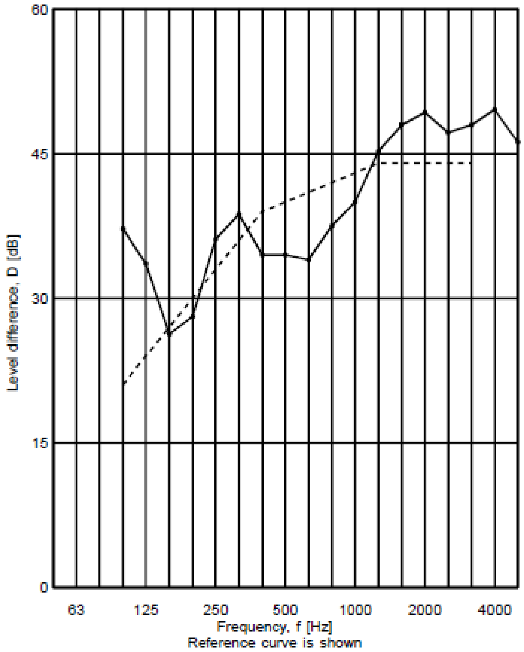
Appendix B – Transmission noise of Spiralflow

Laboratory measurement of airborne sound insulation of single element DS/EN ISO 10140-2:2010

Project: 56195
Date of test: 18 May 2018
Test location: DTU Building 355
Source room: 004
Receiving room: 003
Test object: Spiral flow ventilation unit prototype

Partition area, S: 0.2 m²
Receiving room volume: 215 m³

Frequency f [Hz]	$D_{n,e}$ 1/3-oktav [dB]
100	37.2
125	33.6
160	26.3
200	28.1
250	36.1
315	38.7
400	34.5
500	34.5
630	34.0
800	37.5
1000	40.0
1250	45.2
1600	48.0
2000	49.3
2500	47.2
3150	48.0
4000	49.6
5000	46.2



Weighted element-normalised level difference in accordance with DS/EN ISO 717-1:2013:

$D_{n,e,w}$ (C; C_{tr}) = 40 (-1;-3) dB

Based on laboratory measurements in accordance with DS/ISO 10140:2-2016

Conducted by David Duhalde Rahbæk
Akustisk Teknologi, DTU

Appendix C – Sound power of Spiralflow

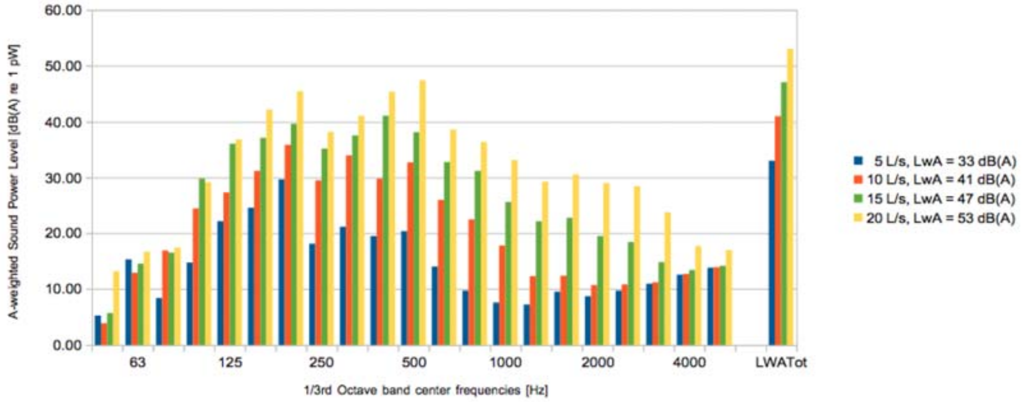


Figure 30 - Indoor sound power of the SpiralFLow unit supply fan at air flows 5, 10, 15, and 20 L/s. Total sound power is seen in the legend for each setting. Calculation based on investigative measurement.

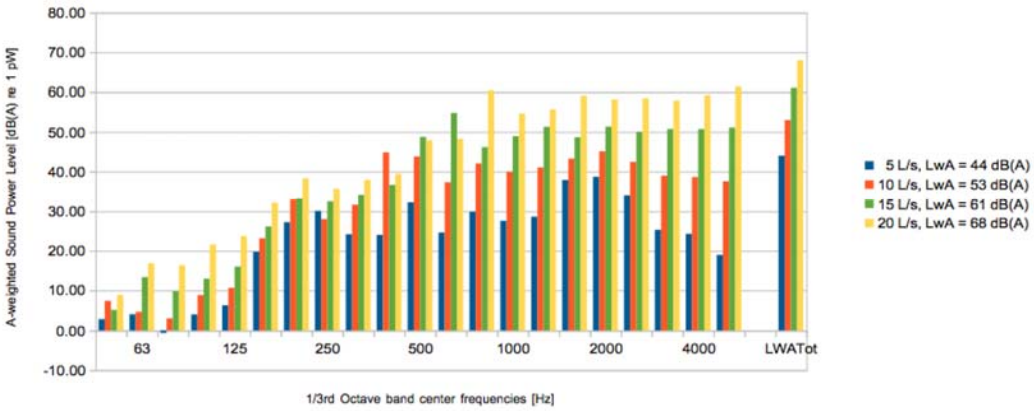


Figure 31 - Indoor sound power of the SpiralFLow unit exhaust fan at air flows 5, 10, 15, and 20 L/s. Total sound power is seen in the legend for each setting. Calculation based on investigative measurement.

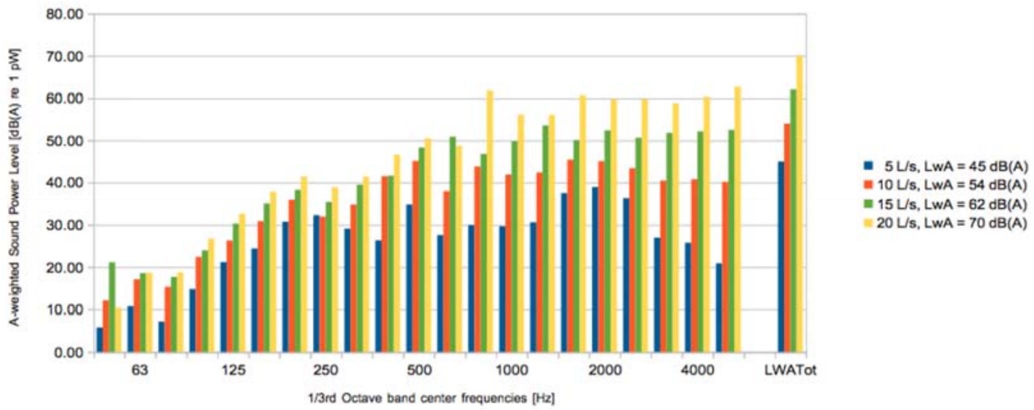


Figure 32 - Indoor sound power of the SpiralFLow unit with both fans at air flows 5, 10, 15, and 20 L/s. Total sound power is seen in the legend for each setting. Calculation based on investigative measurement.

Outdoor sound power

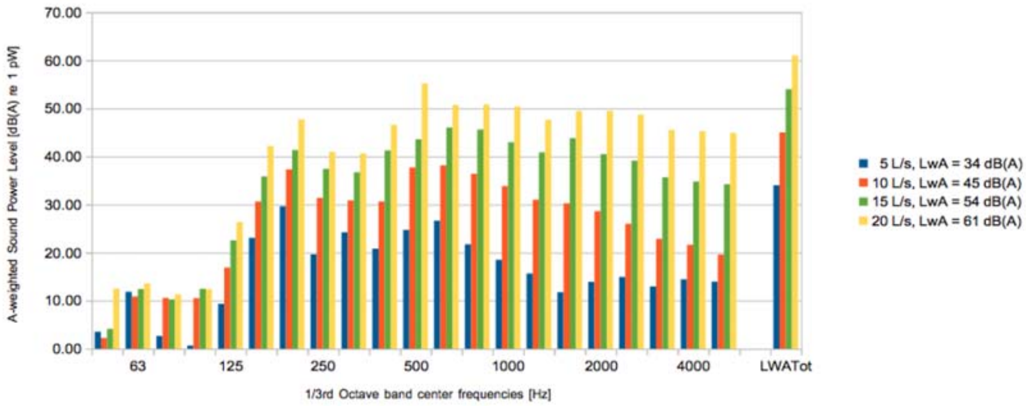


Figure 33 - Outdoor sound power of the SpiralFLow unit supply fan at air flows 5, 10, 15, and 20 L/s. Total sound power is seen in the legend for each setting. Calculation based on investigative measurement.

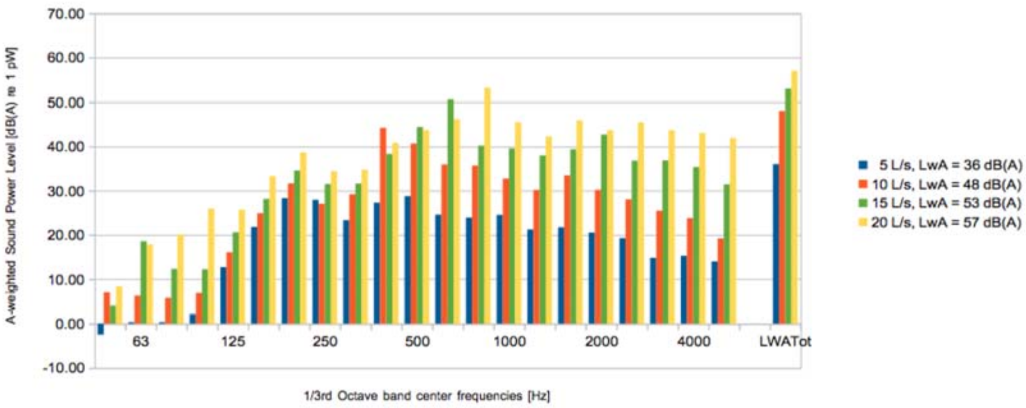


Figure 34 - Outdoor sound power of the SpiralFLow unit exhaust fan at air flows 5, 10, 15, and 20 L/s. Total sound power is seen in the legend for each setting. Calculation based on investigative measurement.

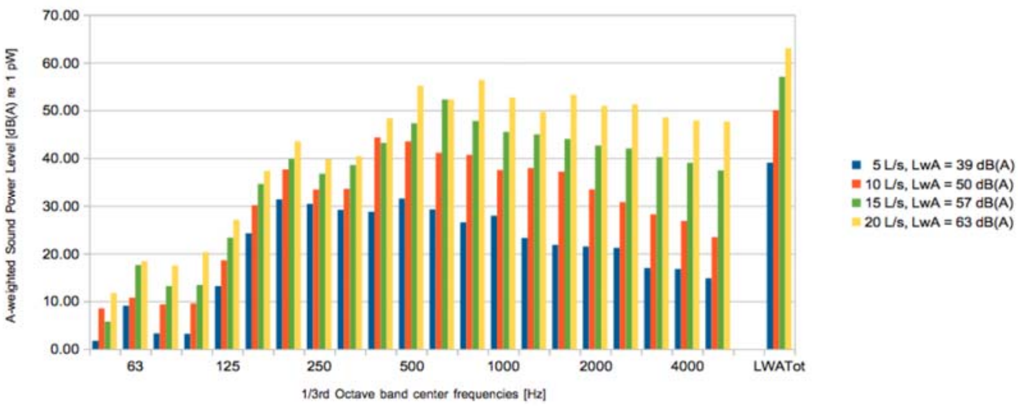
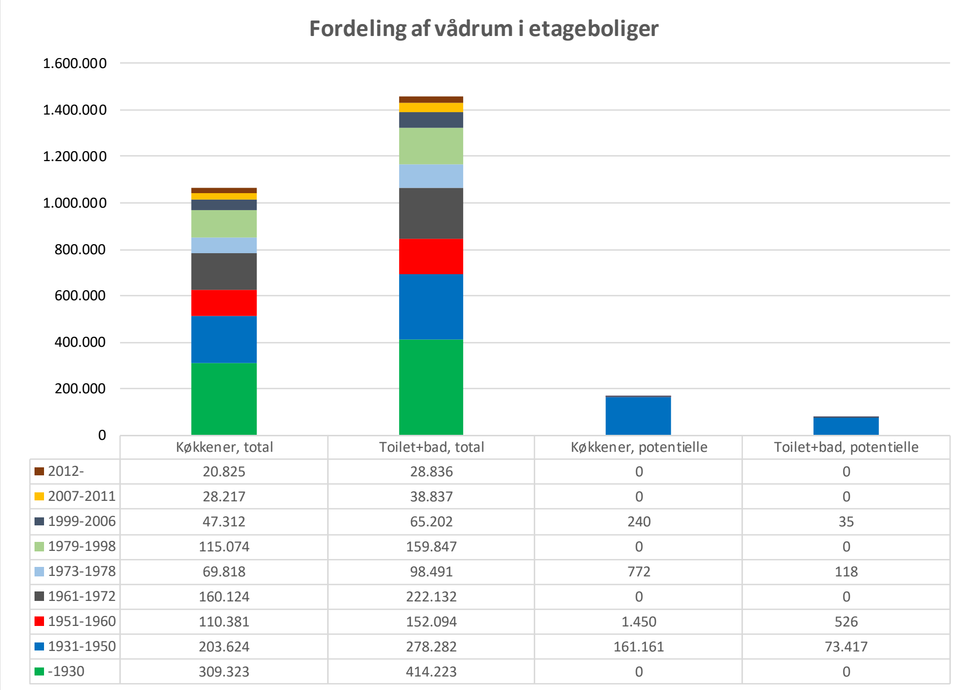


Figure 35 - Outdoor sound power of the SpiralFLow unit with both fans at air flows 5, 10, 15, and 20 L/s. Total sound power is seen in the legend for each setting. Calculation based on investigative measurement.

Appendix D – Overview of the market estimation tool

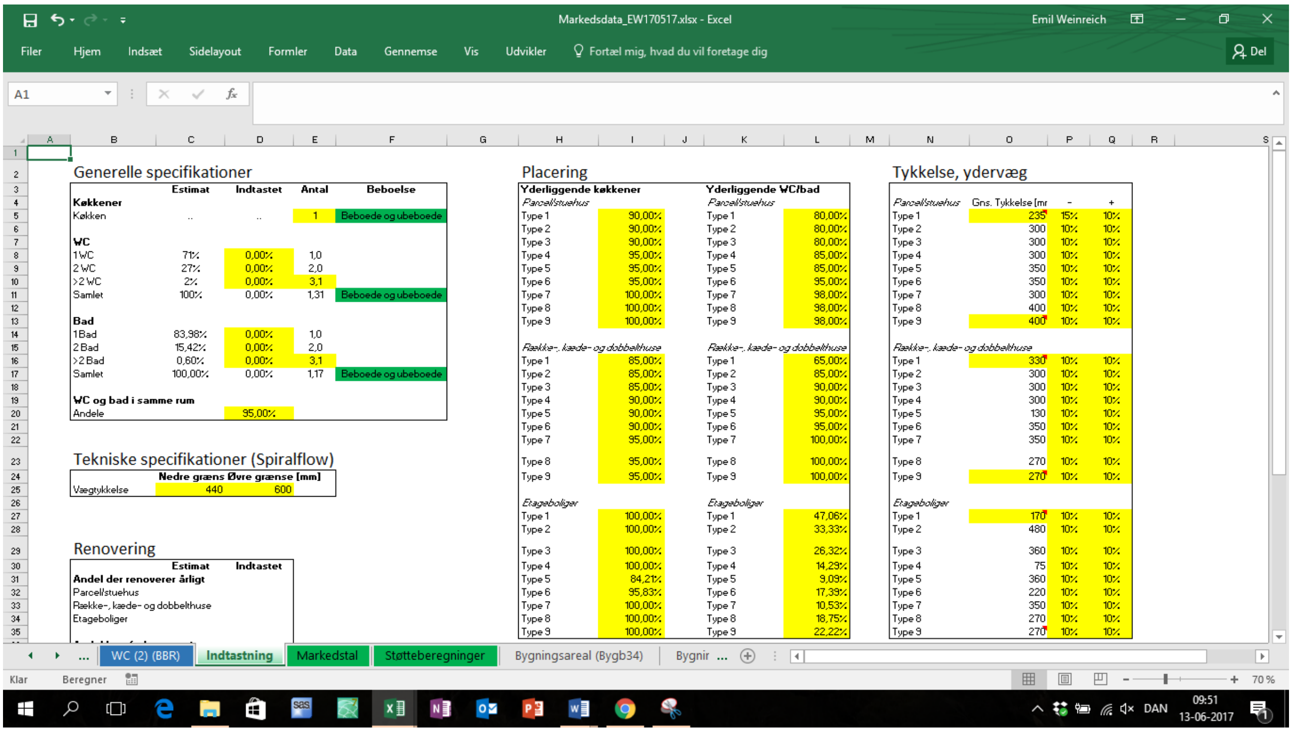
Market example

If Spiralflow could be used for wall thicknesses from 440mm to 600mm. The potential market would then be roughly 164.000 kitchens and 74.000 toilets and bathrooms. The majority of these are found in housings from the period 1931-1950, because of the wall thickness these buildings has.

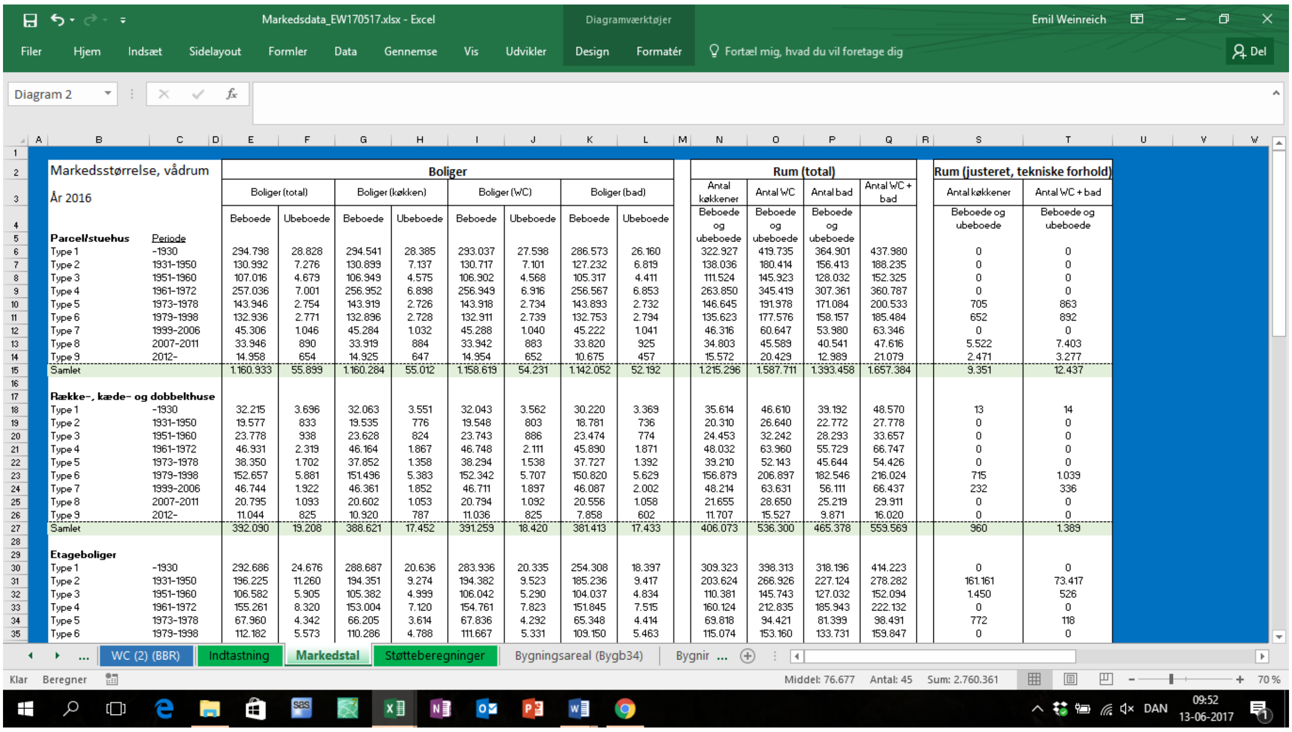


	Køkkener, total	Toilet+bad, total	Køkkener, potentielle	Toilet+bad, potentielle
2012-	20.825	28.836	0	0
2007-2011	28.217	38.837	0	0
1999-2006	47.312	65.202	240	35
1979-1998	115.074	159.847	0	0
1973-1978	69.818	98.491	772	118
1961-1972	160.124	222.132	0	0
1951-1960	110.381	152.094	1.450	526
1931-1950	203.624	278.282	161.161	73.417
-1930	309.323	414.223	0	0

Input of variables



Market numbers



Markedsstørrelse, vådrum		Boliger								Rum (total)				Rum (justeret, tekniske forhold)	
År 2016		Boliger (total)		Boliger (køkken)		Boliger (WC)		Boliger (bad)		Antal køkkener	Antal WC	Antal bad	Antal WC + bad	Antal køkkener	Antal WC + bad
		Beboede	Ubeboede	Beboede	Ubeboede	Beboede	Ubeboede	Beboede	Ubeboede	Beboede og ubeboede	Beboede og ubeboede	Beboede og ubeboede		Beboede og ubeboede	Beboede og ubeboede
Parcel/stuehus	Periode														
Type 1	-1930	294.798	28.828	294.541	28.385	293.037	27.598	286.573	26.160	322.927	419.735	364.901	437.980	0	0
Type 2	1931-1950	130.992	7.276	130.899	7.137	130.717	7.101	127.232	6.819	138.036	180.414	156.413	188.235	0	0
Type 3	1951-1960	107.016	4.679	106.949	4.575	106.902	4.568	105.317	4.411	111.524	145.923	128.032	152.325	0	0
Type 4	1961-1972	257.036	7.001	256.952	6.898	256.949	6.916	256.567	6.853	263.850	345.419	307.361	360.787	0	0
Type 5	1973-1978	143.946	2.754	143.919	2.726	143.918	2.734	143.893	2.732	146.645	191.978	171.084	200.533	705	863
Type 6	1979-1998	132.936	2.771	132.896	2.728	132.911	2.739	132.753	2.794	135.623	177.576	158.157	185.484	652	892
Type 7	1999-2006	45.306	1.046	45.284	1.032	45.288	1.040	45.222	1.041	46.316	60.647	53.980	63.346	0	0
Type 8	2007-2011	33.946	890	33.919	884	33.942	883	33.820	925	34.803	45.589	40.541	47.616	5.522	7.403
Type 9	2012-	14.958	654	14.925	647	14.954	652	10.675	457	15.572	20.429	12.989	21.079	2.471	3.277
Samlet		1.160.933	55.899	1.160.284	55.012	1.158.619	54.231	1.142.052	52.192	1.215.296	1.587.711	1.393.458	1.657.384	9.351	12.437
Række-, kæde- og dobbelthuse															
Type 1	-1930	32.215	3.696	32.063	3.551	32.043	3.562	30.220	3.369	35.614	46.610	39.192	48.570	13	14
Type 2	1931-1950	19.577	833	19.535	776	19.548	803	18.781	736	20.310	26.640	22.772	27.778	0	0
Type 3	1951-1960	23.778	938	23.628	824	23.743	886	23.474	774	24.453	32.242	28.293	33.657	0	0
Type 4	1961-1972	46.931	2.319	46.164	1.867	46.748	2.111	45.890	1.871	48.032	63.960	55.729	66.747	0	0
Type 5	1973-1978	38.350	1.702	37.852	1.358	38.294	1.538	37.727	1.392	39.210	52.143	45.644	54.426	0	0
Type 6	1979-1998	152.657	5.881	151.496	5.383	152.342	5.707	150.820	5.629	156.879	206.897	182.546	216.024	715	1.039
Type 7	1999-2006	46.744	1.922	46.361	1.852	46.711	1.897	46.087	2.002	48.214	63.631	56.111	66.437	232	336
Type 8	2007-2011	20.795	1.093	20.602	1.053	20.794	1.092	20.556	1.058	21.655	28.650	25.219	29.911	0	0
Type 9	2012-	11.044	825	10.920	787	11.036	825	7.858	602	11.707	15.527	9.871	16.020	0	0
Samlet		392.090	19.208	388.621	17.452	391.259	18.420	381.413	17.433	406.073	536.300	465.378	559.569	960	1.389
Etageboliger															
Type 1	-1930	292.686	24.676	288.687	20.636	283.936	20.335	254.308	18.397	309.323	398.313	318.196	414.223	0	0
Type 2	1931-1950	196.225	11.260	194.351	9.274	194.382	9.523	185.236	9.417	203.624	266.926	227.124	278.282	161.161	73.417
Type 3	1951-1960	106.582	5.905	105.382	4.999	106.042	5.290	104.037	4.834	110.381	145.743	127.032	152.094	1.450	526
Type 4	1961-1972	155.261	8.320	153.004	7.120	154.761	7.823	151.845	7.515	160.124	212.835	185.943	222.132	0	0
Type 5	1973-1978	67.960	4.342	66.205	3.614	67.836	4.292	65.348	4.414	69.818	94.421	81.399	98.491	772	118
Type 6	1979-1998	112.182	5.573	110.286	4.788	111.667	5.331	109.150	5.463	115.074	153.160	133.731	159.847	0	0